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EFFICIENCY TEST OF TWO 22,500 H.P. SINGLE REDUCTION GEAR UNITS BY METHOD OF LOSSES.

BY LIEUT. COMDR. H. G. DONALD, U. S. N., MEMBER.

INTRODUCTION.

A most interesting, thorough and instructive efficiency test of two large single reduction, double helical gear units was recently conducted by the Falk Company of Milwaukee, Wisconsin. The data was scientifically and carefully collected and is considered most reliable. It shows conclusively that we may expect 98½ per cent mechanical efficiency from mechanical single reduction gears of large power.

The primary object of the test was for the purpose of demonstrating the condition of noise in the gearing when operated at full load and speed, but in carrying out this test the Falk Company took the opportunity of obtaining the valuable data as set forth later. It will be of interest to state that the condition of noise at full load, full speed was most satisfactory. Two persons were able to converse in an ordinary tone of voice when six feet apart and about two feet clear of the gear case. There was no sound of objectionable pitch which might lend to confusion in an engine room of a vessel. The vibration was not noticeable at any speed up to 480 R.P.M. These facts point to the high grade of workmanship used and the exactness with which the hobbing of the gearing was carried out.

The gearing tested consisted of two sets of double helical gear units for Light Cruisers 7 and 8 (The U. S. S. *Raleigh* and U. S. S. *Detroit,* respectively), now building at the Fore River plant of the Bethlehem Shipbuilding Corporation. The following is a brief description of the machinery for main propulsion and a full description of the gearing.

DESCRIPTION OF GEAR UNITS.

The power is divided on four propeller shafts, each of which is driven through gearing by one high pressure turbine and one intermediate pressure turbine on the first pinion, and one low pressure turbine of the double flow type and one astern turbine on the second pinion. The ahead power is rated at 22,500 S.H.P. per propeller shaft, although the turbines will be capable of developing slightly greater than 25,000 S.H.P. per shaft. The astern power is sufficient to drive the vessel astern at twenty knots.

Each propeller shaft has one set of reduction gears which are single reduction, double helical, data as follows:

R.P.M. propeller shaft	400
R.P.M. turbine (all) shaft	2513
Speed reduction	6.28 to 1
Pitch diameter, gear, inches	84.286
Root diameter, gear, inches	83.772
Pinion, pitch diameter, inches	13.429
Pinion, root diameter, inches	12.915
Pinion, number of teeth	47
Weight of gear, shaft, pinions and couplings, 1 unit, pounds	30,150
Circular pitch	.8976
Diametral pitch	3½
Pressure angle, degrees	25
Spiral angle, degrees	30
Total length of tooth face, each pinion, inches	48
Distance between helices, inches	22
Outside ends of teeth chamfered	Yes

The material of and physical properties of the gear, pinions and torsion shafts are as follows:

	Gear	Pinion	Torsion shaft
Material	C. S.	Chr. Van. St. Forg.	An. Steel Forg.
Minimum tension stress, pounds	70,000	100,000	80,000
Minimum elastic limit	35,000	75,000	50,000
Elongation in 2 inches, per cent	20	21	25

The main gear is built up of two cast steel gears, rim and spider cast in one piece, and a drum, separating the two gears or helices, is bolted to each. The low speed shaft, on which the main gears are keyed, carries the main thrust collar at its after end. The main thrust bearing seating is a part of the bed plate for the gear casing.

There are two pinions, located one on each side of the main gear in a horizontal plane through the main gear shaft. Each pinion shaft is eight feet and 10¾ inches long centrally bored to a diameter of 6.500 inches. The pinion is driven through a flexible pin coupling from the after end by means of a solid torsion shaft 6.375 inches in diameter and 10 feet 11⅛ inches long which bolts solidly to the after end turbine shaft, passes through the pinion and has secured to the after end its half of the flexible pin coupling. The torsion shaft is supported by a bearing at the after end and at about its mid length a 2-inch wide collar is provided which fits snugly in the pinion shaft to prevent undue whipping of the torsion shaft. A special feature of the pinions, whereby noisy operation is greatly reduced is that each helix is undercut with an annular opening, ¾ inch on the radius, for a distance of 10 inches from each end. This annular opening is packed with asbestos and sealed with a steel ring. Not only is noise deadened, but also, in this manner, a more even tooth pressure is maintained for the length of the helix.

Lubrication of the teeth is provided by palm-shaped sprayers, four per helix or eight per pinion, located for ahead running only.

Each pinion has four bearings, diameter 9.250 inches, two of which are 10 inches long, and the other two are 9 inches long. The main gear has two bearings, 15 inches in diameter and 12 inches long. The torsion shaft has one bearing 9.250 inches diameter and 6 inches long. The bearing metal is babbitt composition, copper 4, tin 89½, antimony 7½. For the pinion and torsion shaft bearings, oil is fed in at the side of all bearings, with a reservoir at each side ¾ inch in circumferential width, and which has at each end two 3/16 × 3/32 inches deep slots for bleeding off oil. For the main gear bearings, the reservoir is 4 inches in circumferential width and has at each end two slots ¼ × 3/32 inches deep for bleeding off oil.

The bearing pressures are as follows:

	Pounds, Square Inches.
Inboard pinion bearings	136.5
Outboard pinion bearings	125.0
Gear bearings	70.8

The base of the gear case is of cast iron, outside dimensions 11 feet 2 inches wide by 11 feet 6 inches fore and aft. The oil pan is of sheet metal. The upper half of the gear case consists of a cast iron frame on which is built up a double sheet metal cover, all joints welded, the space between the two covers being filled with asbestos for reducing the noise of the gearing.

SET UP FOR TEST.

The test of the gear units was conducted at no load, one-half load and full load, and at varying speeds under each condition of loading to demonstrate the condition of noise and also to determine the efficiency of the gear units under the above varying conditions. The viscosity of the oil was varied from 250 to 1000 seconds saybolt to ascertain the effect of varying the viscosity of the lubricant upon mechanical efficiency of gear drives.

The loading of the gears was accomplished by using the "Torsion shaft method" as described under "Method of Conducting Test."

Two identical gear units, as described above, were lined up on parallels, bolted to embedded semi-steel bed plates. The gear cases were set forward end to forward end and the low speed (main gear) shafts were joined with a specially constructed flexible pin coupling. The respective torsion shaft flanges of the units were connected by auxiliary torsion shafts carried in separate bearings and solidly bolted together.

In order to keep perfect alignment and to reduce friction losses between the torsion coupling halves to a minimum when the load was being applied to the teeth, as described later, the ends of the torsion shafts were allowed to move freely in ball bearings.

The torsion shafts had been designed for a full load torsional stress of only 7600 pounds in order to insure a very comfortable factor of safety below their elastic limit.

The two gear units were driven by one D.C. motor, 1100 H.P., 250 volts, 400 R.P.M., through a flexible pin coupling connecting to the propeller end coupling on the low speed shaft of one of the units.

The oil for lubrication was supplied by two pumps taking suction from a 1000-gallon oil drain tank and discharging to the bearings and gear sprayers through eight coolers of Falk Company design and one Griscom-Russell multi-whirl cooler all arranged in parallel. At full load and full speed these coolers absorbed about 75 per cent of the heat generated by friction in the gear units, the remaining 25 per cent being dissipated by radiation, conduction, etc. Meters were provided for measuring the oil flow through each cooler and the total oil through the pumps. Gauges were installed for taking the pressure of the oil at pump discharge, when leaving the coolers and at each bearing and gear sprayer. Thermometers were provided for recording the temperature of circulating water injection, and discharge from each cooler, as well as oil temperatures to and from each cooler and from each bearing and gear drain.

The speed of the units was taken by a geared counter supplemented by an electric revolution indicator for the purpose of quickly obtaining approximate speed desired.

The speed of the driving motor was controlled by means of a water rheostat, at the same time observing the revolution counter and indicator. The motor losses were carefully determined. The armature resistance and heat losses are plotted on Plate I and the stray power losses are shown on Plate II.

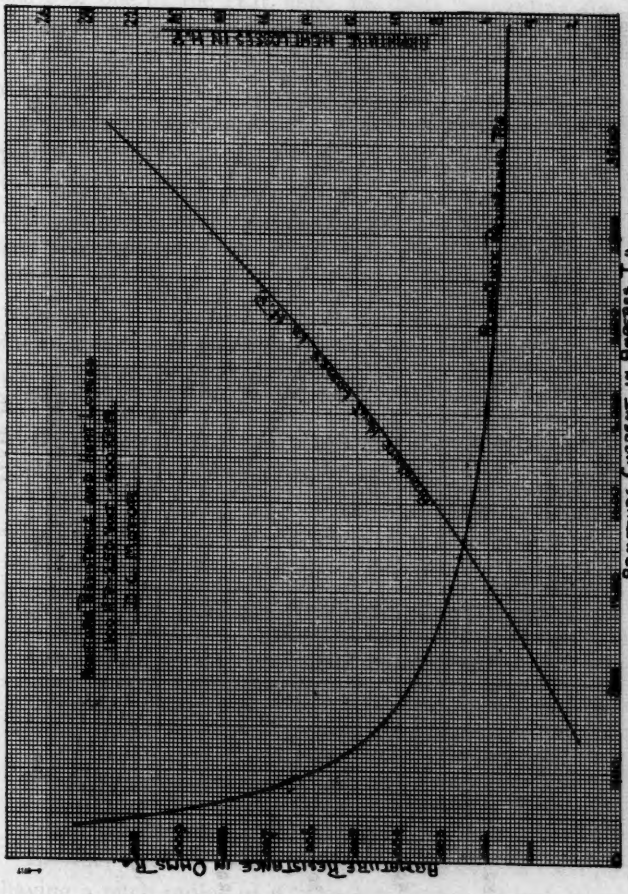


PLATE I.

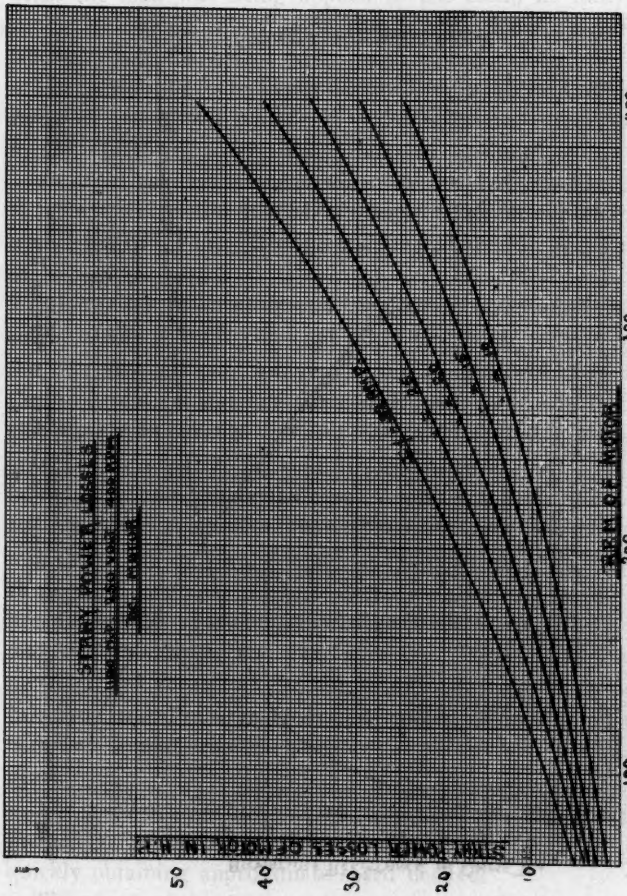

STRAY POWER LOSSES
WHEN DRIVING FROM MOTOR
STRAY POWER LOSSES OF MOTOR IN H.P.
0
10
20
30
40
50
100
200
300
400
SPEED OF MOTOR

PLATE II.

METHOD OF CONDUCTING TEST.

The motor having been connected to the end of the low speed shaft of one of the gear units, the bolts, connecting the two halves of the torsion couplings connecting each set of pinions were loosened and a stop applied to one of each of the pairs of coupling flanges, thus preventing rotation. Specially designed wrenches of 6 feet leverage were then applied to the free companion half coupling and weighted so as to transmit by torque, through the twisted torsion shafts, a pressure on the pinion and main gear teeth equivalent to that obtained at the desired H.P. and given R.P.M.

Care was taken at this time that the adjoining coupling faces were slightly separated to allow a free relative movement of the two halves of the two couplings (one on each side). The pin couplings which connect the torsion shafts with the pinions, had plenty of play to take care of any variation in the length of the torsion shafts due both to the twisting when torque was applied and also to temperature changes.

In order to correct for the fact that when the levers were weighted they took a slight angle to the horizontal, the projected length in the horizontal plane was calculated and the weight adjusted according to the law of movements.

A sample calculation will more fully explain the method:—

When each of the two units develops 22,500 S.H.P. or 11,250 S.H.P. per pinion with a speed of 400 R.P.M., the pressure P in pounds on each pinion is—

$$P = 33{,}000 \times 11{,}250 \times \frac{12}{3.1416 \times 84.285 \times 400} = 42{,}000 \text{ lbs.}$$

84.285 is the pitch diameter of the main gear.

In order to exert such a pressure on the teeth of each pinion, having a pitch radius of 6.71 inches, with a 72-inch lever, the lever must be weighted, when in the horizontal plane, with a weight of—

$$\frac{42{,}000 \times 6.71}{72} = 3{,}920 \text{ pounds.}$$

The stops prevented the gears from turning while the load was being applied; but the disconnected torsion shaft coupling halves allowed the torsion shafts, pinions and low speed shafts to be twisted under the heavy stress. The potential energy thus stored in these shafts was at once transmitted to the gear teeth, producing the desired tooth pressure, the force action in one set of gears being counteracted by an equal reaction in the other set.

Having applied the desired tooth pressure load, the torsion shaft couplings were securely bolted together, the bolt holes being elongated to permit of this in any relative position of the coupling halves; a straight chisel mark was made across the periphery of the torsion coupling halves to preserve the record of the strained position of the torsion shafts and to be used as a check after the test to assure that no slippage had occurred.

Weights, wrenches and stops were then removed and the units were ready for test. The gears were got underway by winding one end of a steel wire strap around the face of the torsion coupling, leading the other end to the hook of an overhead traveling crane and then walking off the crane. At the same time a torque was applied to the driving motor, and the gear picked up speed quickly and easily.

Now it is obvious that the total losses in the gears and bearings under such running conditions are identically the same as if the gears were transmitting the load to the propellers. The only difference is that the actual *output* is zero, that is all power put into the gear sets by the motor is chargeable to them as friction losses; in other words, this power is the mechanical power loss of the gear when interposed between the turbines and propeller shaft as a means of reducing the necessary high turbine speed to the equally necessary low propeller speed.

The gear losses in each run have been obtained by deducting the motor losses from the total power supplied to the motor.

Runs were made at 100, 200, 300 and 400 R.P.M. At each speed the tooth pressure was adjusted to get data for no load,

½ load and full load. At each speed and each load the viscosity of the oil was varied by using varying temperatures of oil to bearings to test at 250, 500, 750 and 1000 saybolt seconds. The oil pressure at the manifold supplying the bearings, etc., was maintained at 15 pounds pressure.

The tooth pressures necessary to simulate one-half and full load were calculated from data based on Plate III, showing the vessel's estimated performance curves, obtained from the builders, the Bethlehem Shipbuilding Corporation, Quincy, Mass., plant.

The oil used was Texaco Ursa, the characteristics of which are shown on Plate IV. The specific gravity and specific heat are for the oil as actually used and were calculated by the Falk Company. The viscosity, Plate V, is that published for Ursa oil by the Texaco Company.

Plate VI shows the net losses in H.P. per unit versus output in S.H.P. per unit, with revolutions and lubricating oil viscosity varied. These curves are deduced from data taken on total input to the motor, deducting motor losses as shown on Plates I and II and the friction loss for the four auxiliary torsion shaft bearings. The oil drained from these four bearings was lead off through separate drain pipes fitted with thermometers.

DISCUSSION OF TESTS.

According to the curves of Plate III we may expect to have a S.H.P. of 101,350 per vessel at full load and propeller speed of 400 R.P.M. Since there are four gear units per vessel, each unit is expected to transmit 25,337 S.H.P. at that propeller speed. With lubricating oil of 500 saybolt seconds, we see from Plate VI that the net gear losses per unit at this speed and power are 386 S.H.P. The motor losses for this condition (2 units) were 50 H.P. and the loss chargeable to the auxiliary torsion shaft was 22 H.P.

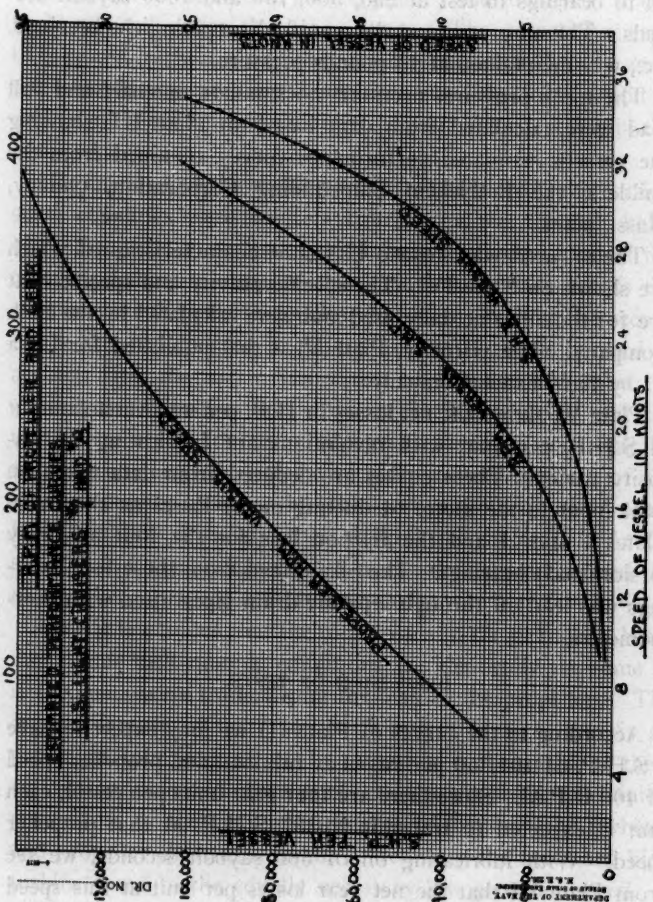

SPEED OF VESSEL IN KNOTS
SPEED OF VESSEL IN KNOTS
SHIP'S TRIAL VESSEL
DR. NO.
SPEED OF VESSEL IN KNOTS

PLATE III.

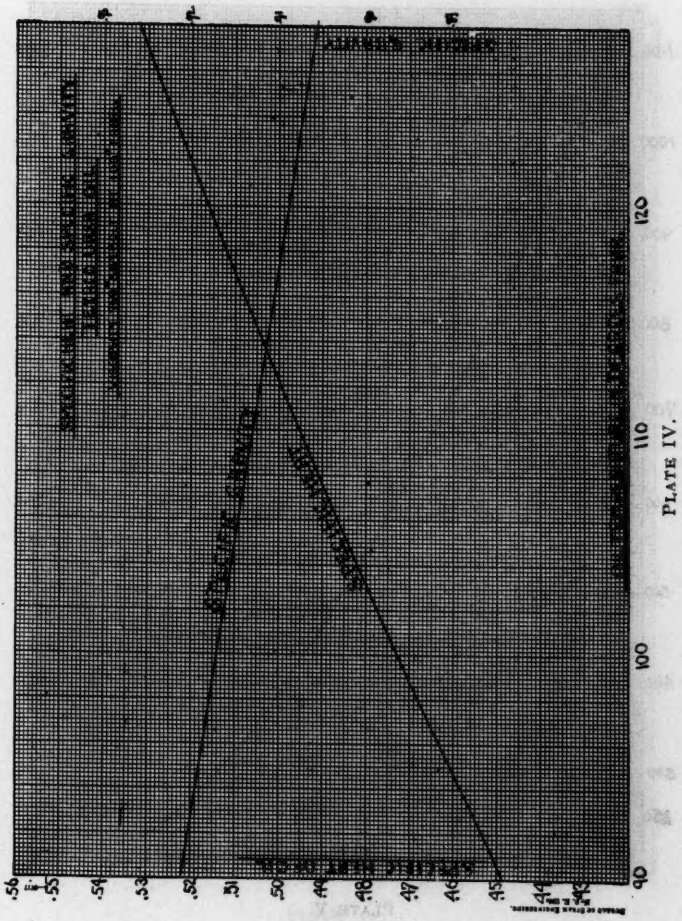


PLATE IV.

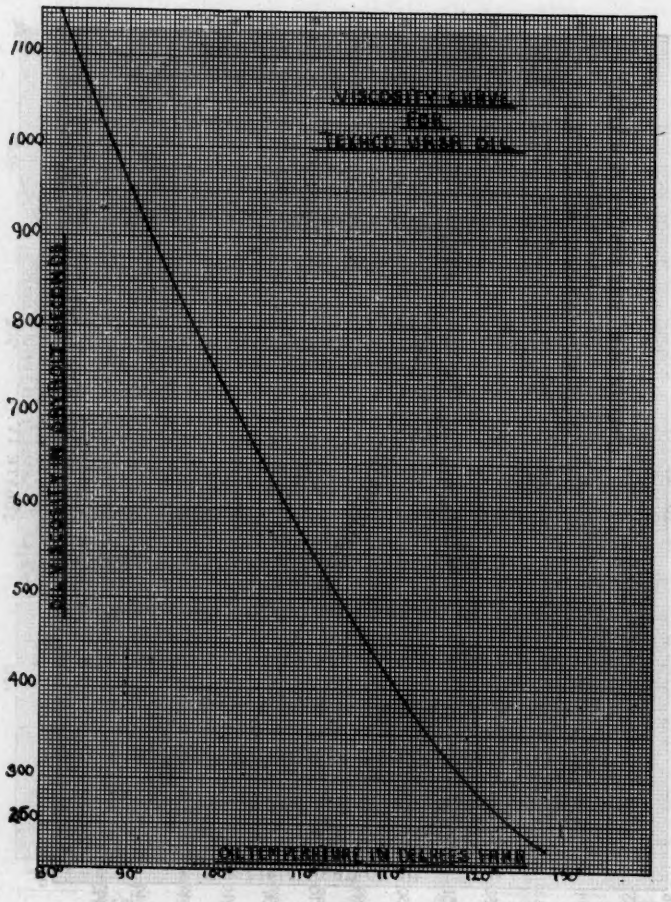

VISCOSITY CURVE
FOR
TEXACO MINOR OIL
SEC. VISCOSITY IN VARYING DEGREES
OIL TEMPERATURE IN DEGREES FAHR.
1100
1000
900
800
700
600
500
400
300
250
80°
90°
100°
110°
120°
130°

PLATE V.

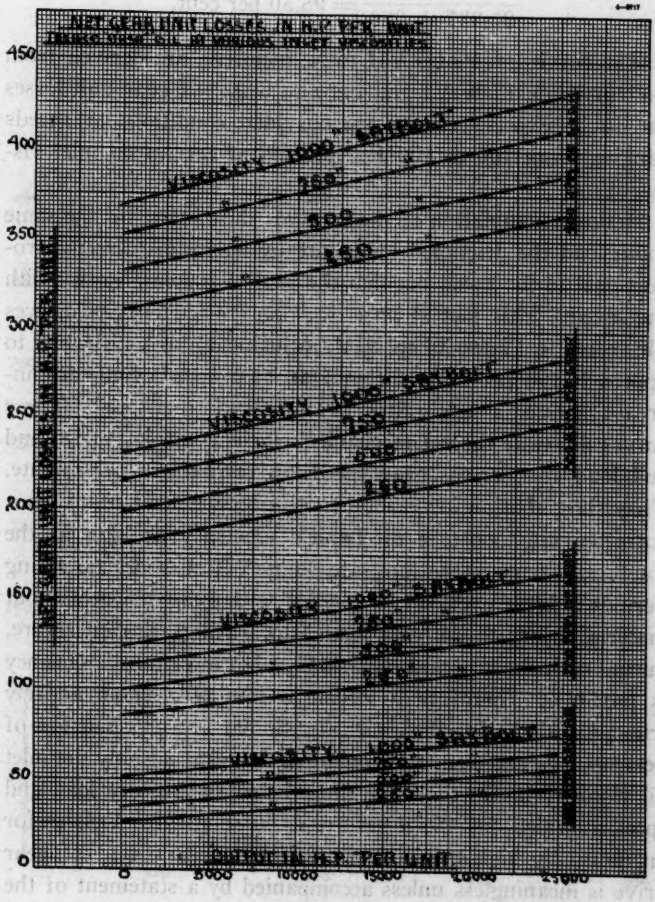


PLATE VI.

The mechanical efficiency of a unit was, therefore—

$$\frac{25{,}337}{25{,}337 + 386} = 98.50 \text{ per cent.}$$

The curves on Plates VII to X, inclusive, are derived from the curves on Plate VI and indicate what net gear unit losses are to be expected under various load conditions at speeds ranging from 100 to 400 R.P.M. and at certain oil inlet viscosities.

A close examination of these curves shows that for the same tooth load the frictional losses are by no means directly proportional to the speed but that the rate of increase grows with the speed. This fact at once explains the tendency of curves illustrated in Plates XI to XIV, inclusive, all of which tend to approach 100 per cent efficiency at "O" speed, while for increase in speed a steady drop in efficiency is registered. These curves illustrate the net mechanical efficiency at ¼, ½, ¾ and full load for various gear speeds in revolutions per minute. They show that an increase in tooth pressure load results in a corresponding boost in mechanical efficiency as long as the total load remains within safe limits. For short operating periods a great overload is possible with gear drives without material injury to the gear teeth and bearings, and therefore, during such short period remarkably high mechanical efficiency is possible. Therefore in order to compare, on an entirely true and fair basis, efficiencies for similar types and makes of gearing, it is necessary to stipulate in the tests not only the inlet oil viscosity but at the same time the tooth pressure load and speed at which the efficiencies are to be obtained. A claim for an unusually high mechanical efficiency for any type of gear drive is meaningless unless accompanied by a statement of the oil viscosity, the tooth load and the speed.

Plate XV is derived from the previous set of curves in Plate XII by cutting them at right angles, using the load at 400 R.P.M. as abscissae instead of the speed in R.P.M. These

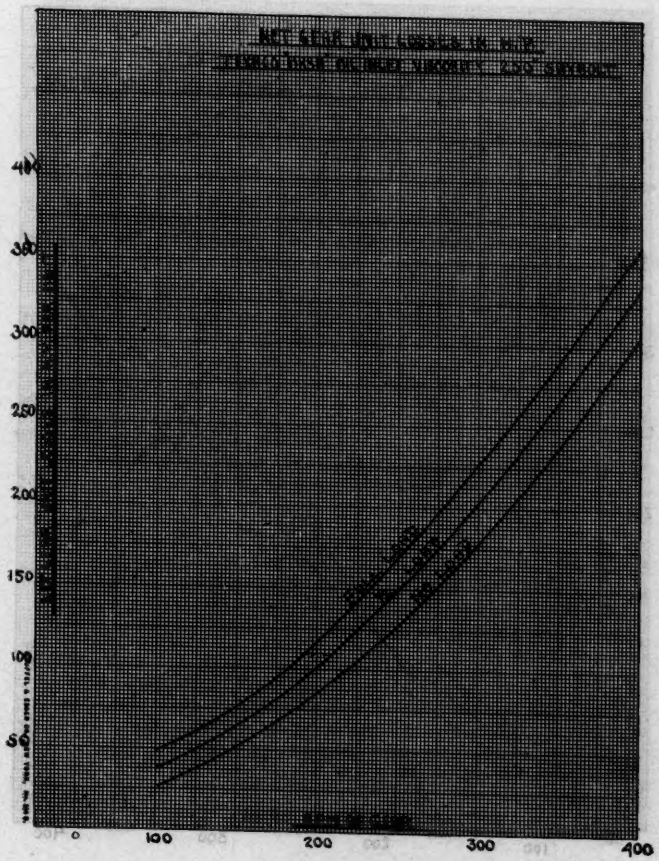


PLATE VII

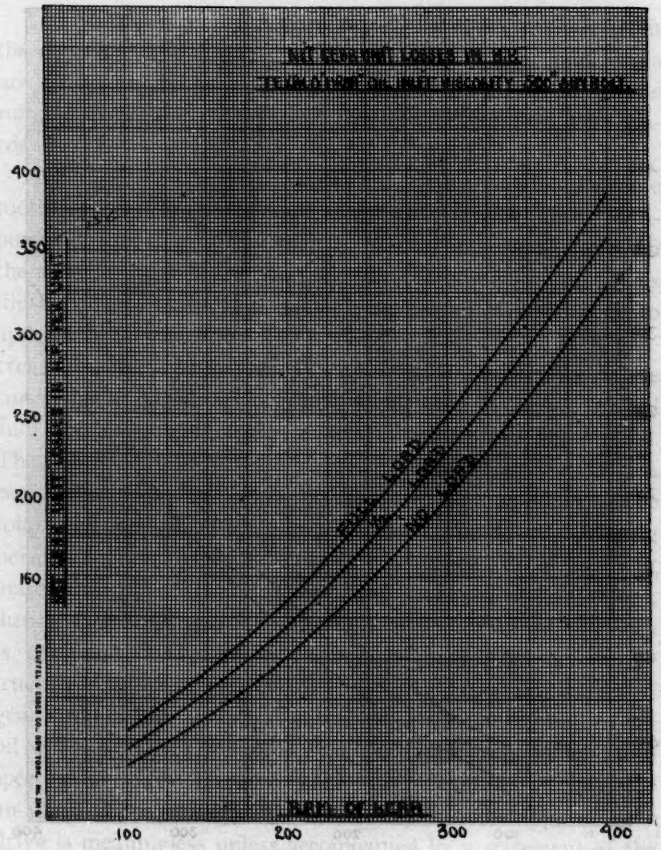


PLATE VIII.

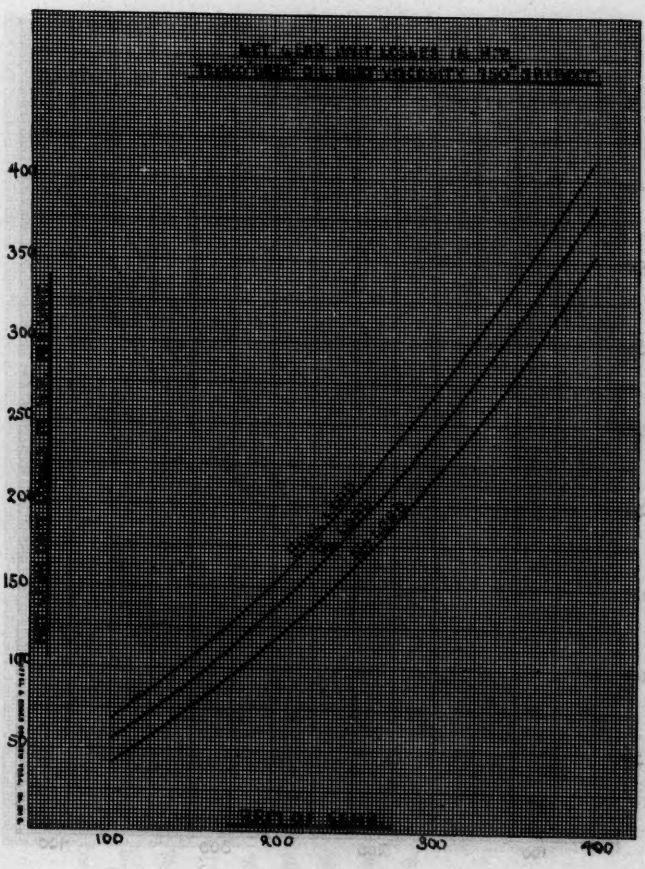


PLATE IX.

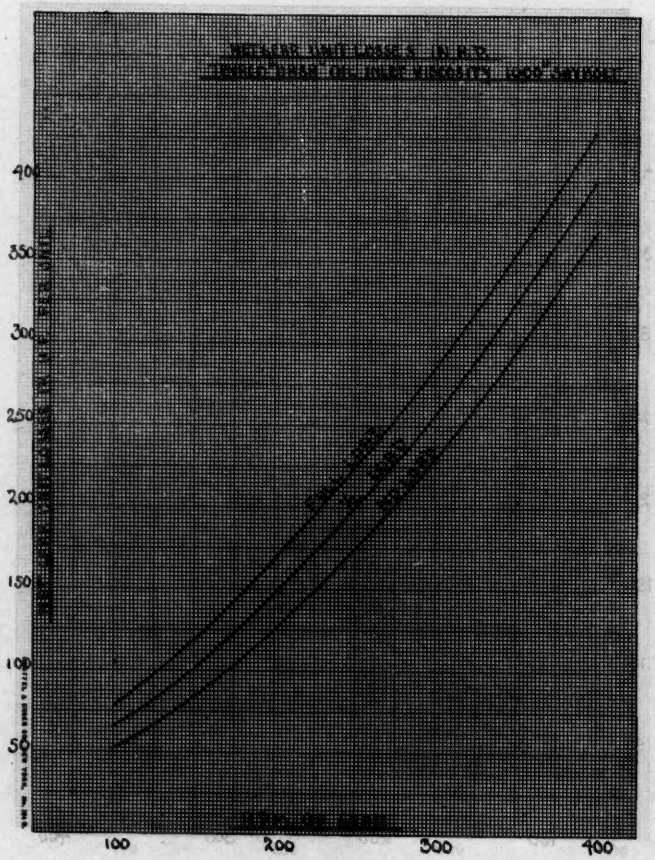


PLATE X.

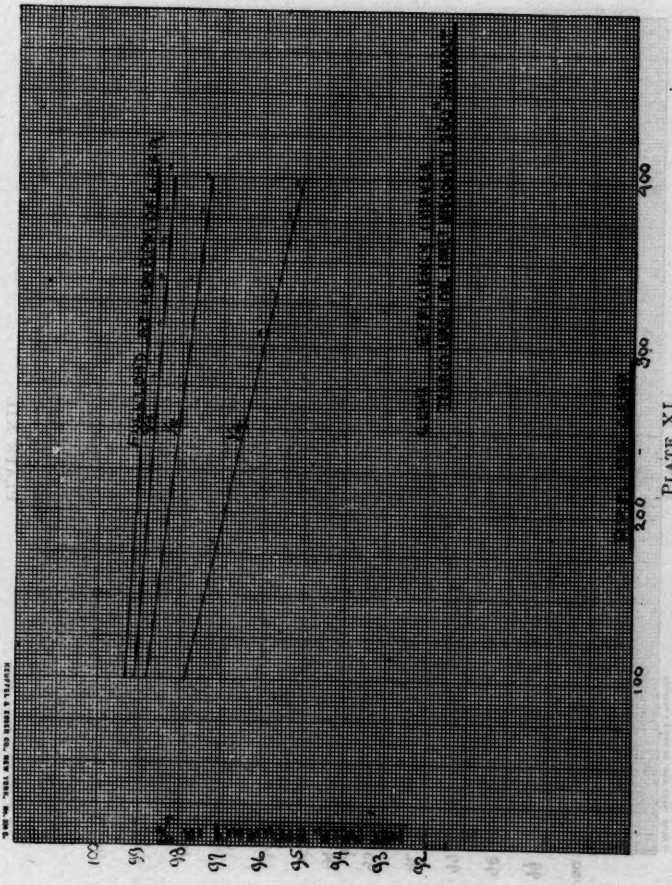


PLATE XI.

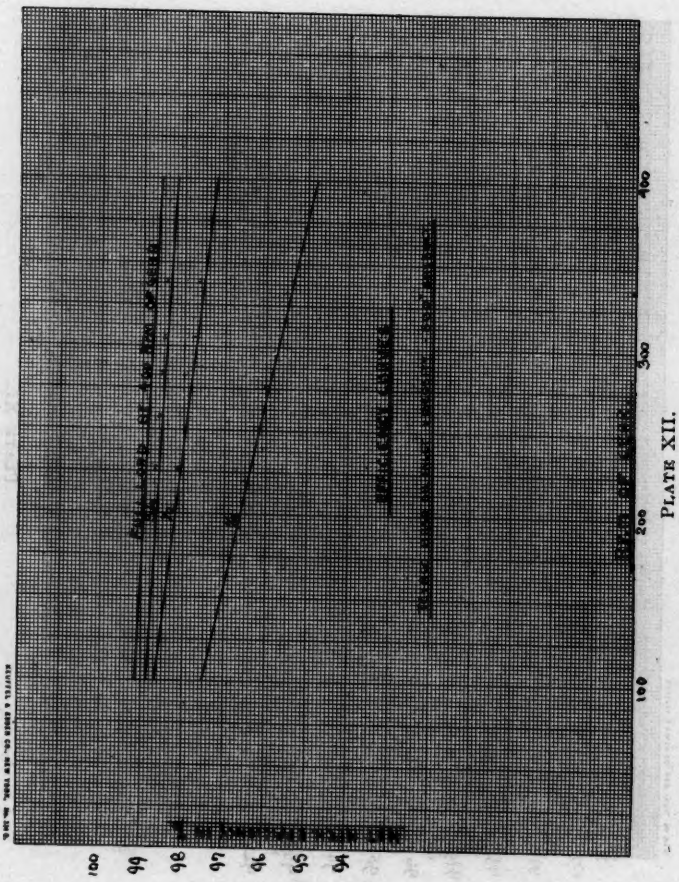


PLATE XII.

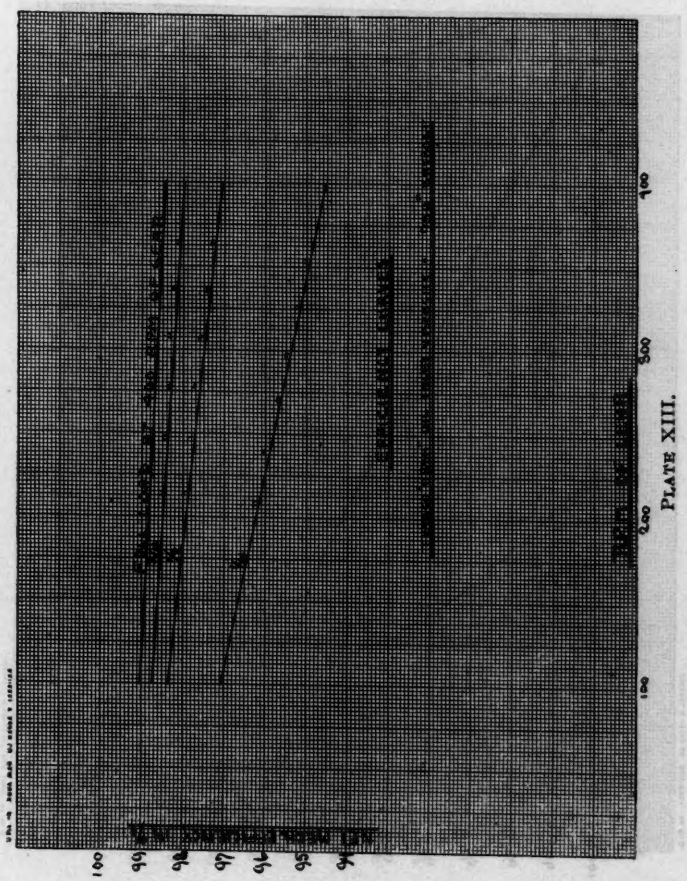
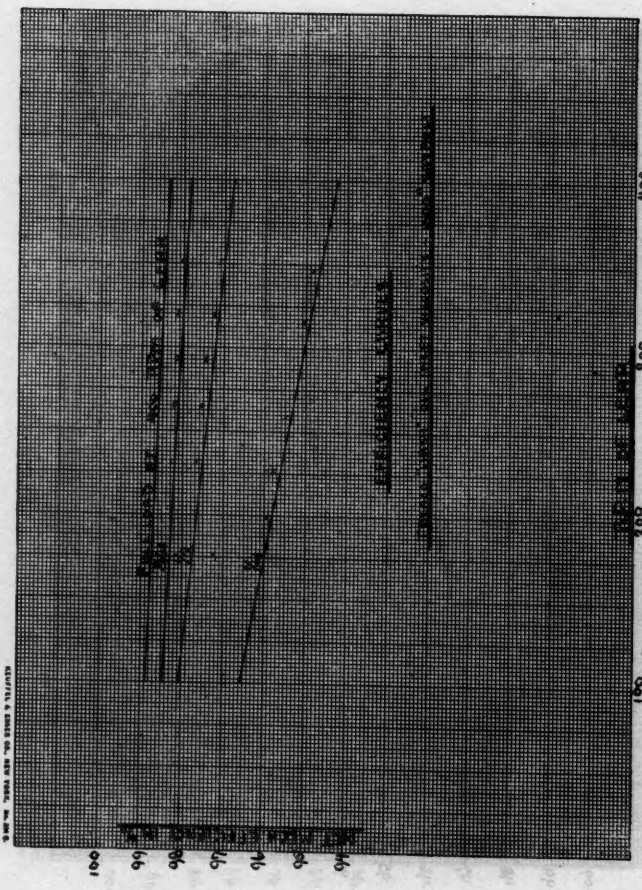


PLATE XIII.


100
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10000

PLATE XIV.

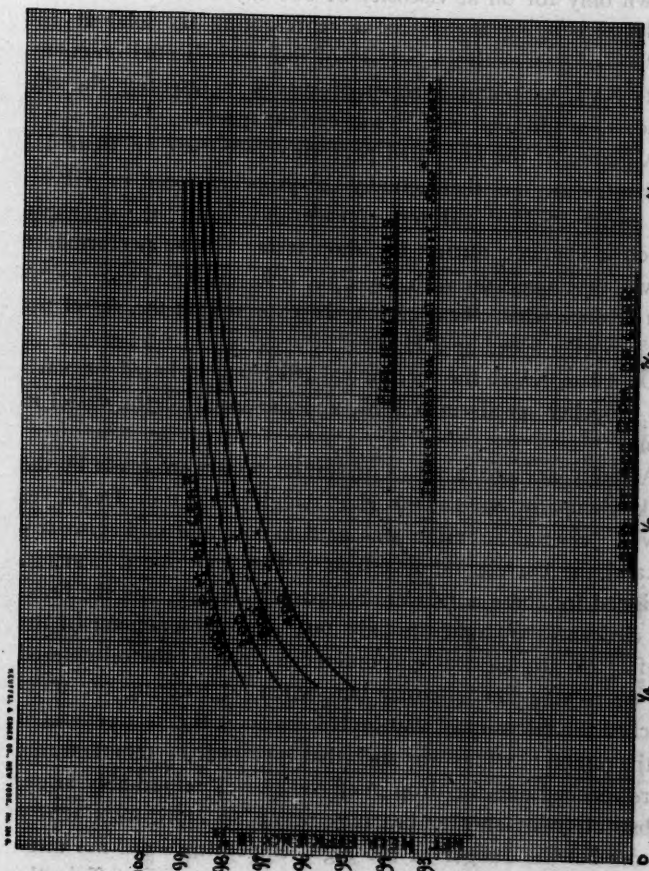


PLATE XV.

curves therefore explain the same results as the previous ones only looked at from a different viewpoint. The curves are drawn only for oil at viscosity of 500 saybolt seconds.

By exchanging the abscissae representing the load at 400 R.P.M. of gear with the S.H.P. of unit as indicated on curve, Plate XVI, we obtain a series of curves representing the efficiencies for different S.H.P.'s at certain speeds.

A glance at the estimated performance curves on Plate III illustrates the fact that actually the propeller speed in R.P.M. for various S.H.P.'s differs considerably from the manner in which it changed during the shop tests. These performance curves in connection with the Falk tests enable one to ascertain at once what efficiency may be expected on the vessel for any propeller speed or S.H.P.

On Plate XVII the mechanical efficiency is based on various propeller speeds in R.P.M. and the speed of the vessel in knots.

A few years ago, a series of similar tests were made by the Falk Company on 14,000 H.P. single reduction Destroyer drives. Special attention was paid to low and high viscosity oil and the corresponding effects. The high speed bearings designed for these units were comparatively long and the bearing clearances amounted to only .001" per inch diameter of the shaft. No trouble was encountered as long as the oil entered the units at a low viscosity but, when oil entering the units reached the viscosity of 750 saybolt seconds, the heat generated in the pinion bearings was such that high bearing temperatures were reached and the drop in efficiency was remarkably large. Substitution of shorter bearings produced more satisfactory results but the danger of too high temperatures did not altogether disappear and the efficiency did not recover sufficiently until bearings with larger clearances were installed.

On the test of the light cruiser gears, at no time was there any manifestation of undue heat in the pinion bearings, not even when the viscosity of the entering of oil was allowed to

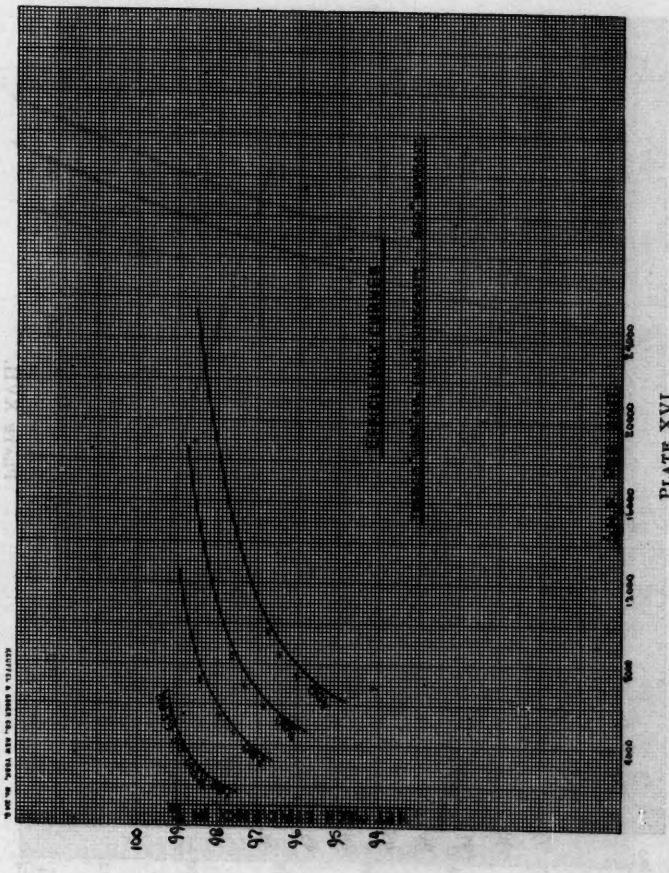


PLATE XVI.

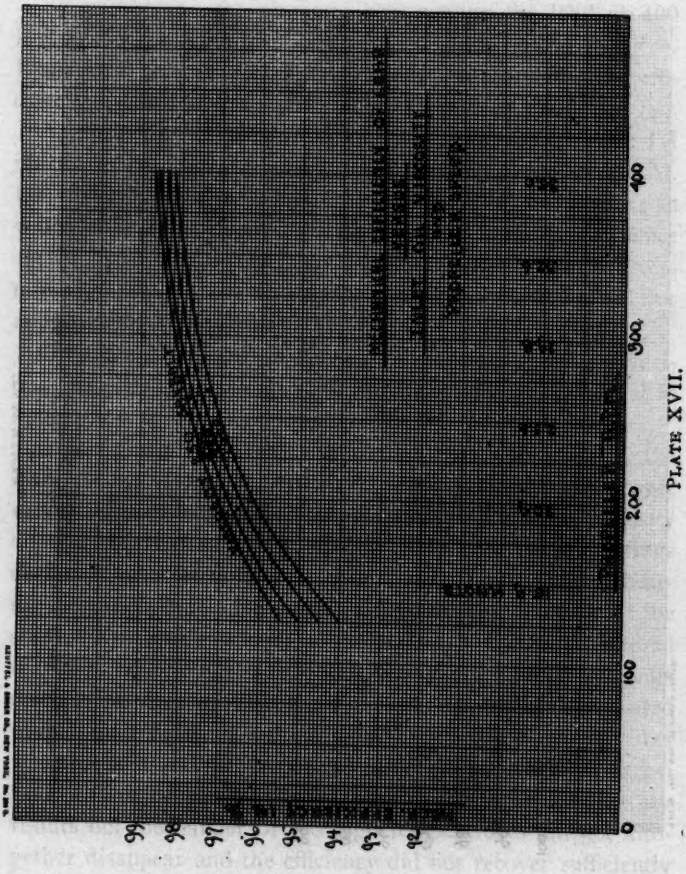


PLATE XVII.

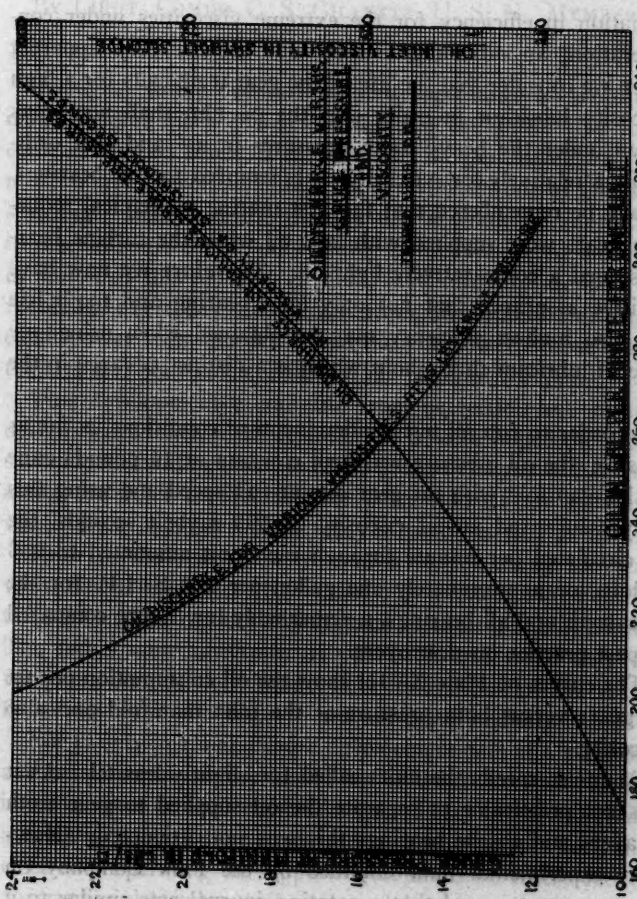


PLATE XVIII.

exceed 1000 saybolt seconds. This proves that the light cruiser bearings are well proportioned and that the bearing clearances are correct. It is for this reason that we find such a small variation in efficiency for the extreme viscosities under consideration, an advantage which will become very apparent when we consider that the life of the gear teeth calls for a high viscosity lubricant in order that the oil film shall not break down under the heavy tooth pressures in use and that ordinarily a relatively low viscosity oil is required for the bearings. It will be remembered that the first marine mechanical reduction gears were supplied with lard oil while the turbine bearings used a comparatively light oil. Therefore we now have positive assurance that not only long bearing and tooth life but also highest efficiency may be obtained when using one type of lubricant throughout which is a distinct advance in the art.

In order to run all tests under identical conditions as far as possible, the oil pressure was kept constant at 15 pounds gage at the manifold with the gear sets at full speed and using this oil pressure and an oil viscosity of 500 saybolt seconds, the oil circulated per minute through each unit amounted to 253 gallons, or approximately one gallon per 100 S.H.P. for the gear teeth and all bearings, a remarkably low figure compared to past practice on destroyers.

The curves of Plate XVIII show the oil consumption per set of gears versus both pressure at the inlet manifold and also viscosity.

At lower speeds the units called for slightly less oil. As it was not practicable to separate the oil supplied to gear teeth sprayers and to the bearings this was not attempted. However, experimental data obtained by the Falk Company on previous tests proved that a rotating journal acts similar to a runner in a centrifugal pump and the higher the surface speed of the journal the more oil flows through the bearing with the same inlet oil temperature and pressure.

MODERN SEARCHLIGHTS.

By Lieut. Comdr. C. S. Gillette, U. S. N., Member.

Though possibly overshadowed in importance by the other many revolutionary changes in fighting equipment that have come as a direct result of the late war, the development of the modern high power searchlight, is nevertheless one of the most startling. It is therefore thought that it would be interesting to discuss some of the outstanding points connected with the evolution of this modern, efficient, high intensity arc with its revolving electrodes, air cooled drum and millions of beam candlepower from our old friend the low power, whose eccentricities and poor efficiency tried our patience for years.

This development is still far from perfection and present designs leave much to be desired. It is to be hoped the necessary impetus will be given to carry present designs still further along the road to the ultimate perfection of an apparatus which finds so many useful fields in peace as well as in war.

Prior to the year 1914 we had struggled along for years with the old low power lights, striving to extend their limitations by various devices and systems of control and operation. The lights were uncertain of operation and when they did function, were of short range, poor finding power, and with a flickering beam of a color to prevent good visibility.

For twenty years and more development was at a standstill, and manufacturers either could not or would not turn out any improvements on the original design. This design, comprising a drum with glass door strips and parabolic mirror, horizontal electrodes operating on supporting columns on tracks moving forward as electrodes were consumed, by magnetically operated palls on threaded rods, appeared in numerous sizes ranging from a nine-inch mirror and a ten-ampère arc to a sixty-inch mirror and a two hundred-ampère arc, or from that suitable

for a small motor boat to the largest fortification type. The arc voltages were from forty-five volts on the smaller size to sixty volts on the larger.

The reasons for the adoption of certain features in the design of these old lights, mainly in regard to electrodes, have never been made clear. The electrodes were of extremely large diameter; a 2¼-inch positive and a 1⅛-inch negative were used to carry 200 ampères, and operated at the freak burning ratio of three parts positive to two parts negative. These proportions suggest a hit and miss design with little thought or money expenditure on research.

The arc voltages and current are likewise very indefinite as to origin. The 200-ampère arc operated at 60 volts, the 120-ampère arc at 60 volts and the 80-ampère arc, having characteristics identical with the higher ampèrage arcs, operated at 45 volts. Experiments on this type of lamp actually proved that no attention had been paid to efficiency, as increased intensities resulted from a deviation from the designed arc voltage and burning ratio of electrodes keeping the original currents. The 200-ampère arc increased its intensity and steady operating characteristics up to a maximum at 95 volts, while the 120-ampère arc and 80-ampère arc operated most satisfactorily at 72 to 75 volts on the old electrode proportion.

This will serve to illustrate the small importance placed on this unit by the makers and the lack of initiative displayed in allowing this condition to continue over a period of twenty years. The first change from the old condition was made early in 1913 when the Navy Department went into the matter and ascertained that the lamps in service could be made more useful and efficient by a simple change in electrode sizes. The burning ratio was changed from 3-2 to 1-1 of positive to negative consumption by length. This single change, using the old current values but generally raising the 45-volt arcs to 50-55 volts, made an increase of twenty to forty per cent in the luminous efficiency of the beam.

This small change was the result of investigation by the Navy of a new type of 44-inch light made by a French concern which used the 1-1 ratio for electrodes. For the same current and voltage as the American light of equal size, this French light produced greater target intensity. The Navy thus took the initiative in the attempt to increase the usefulness of their searchlights. The marked success following, emphasized the already evident fact that this particular ship auxiliary needed attention and the Navy Department was preparing to carry the work further when, in 1914, the entire field was revolutionized by the introduction into the United States of a searchlight of heretofore undreamed of capacities of range, finding power and efficiency.

Heinrich Beck, a German scientist and inventor, requested a test by the Navy Department of his invention, a lighting unit of unique characteristics. He proposed to place at the disposal of the Navy a complete working model of a 44-inch light, ready for ships' service. As the Navy Department was already fully awakened to the fact that its searchlights badly needed development as a defensive and offensive unit in ships' equipment and was contemplating improvements, the offer by Beck was accepted.

This light was sent to the N. Y. Navy Yard for complete test. The result of this test demonstrated beyond doubt that an illuminating agency had been produced in a practical form which in one step had increased the luminous efficiency of the light source three hundred per cent and produced nine times the finding power of the old lights. Coming as it did at the very outbreak of the World War, the vital importance of this design as an adjunct to battleship equipment and for shore and air craft defense was all the more apparent.

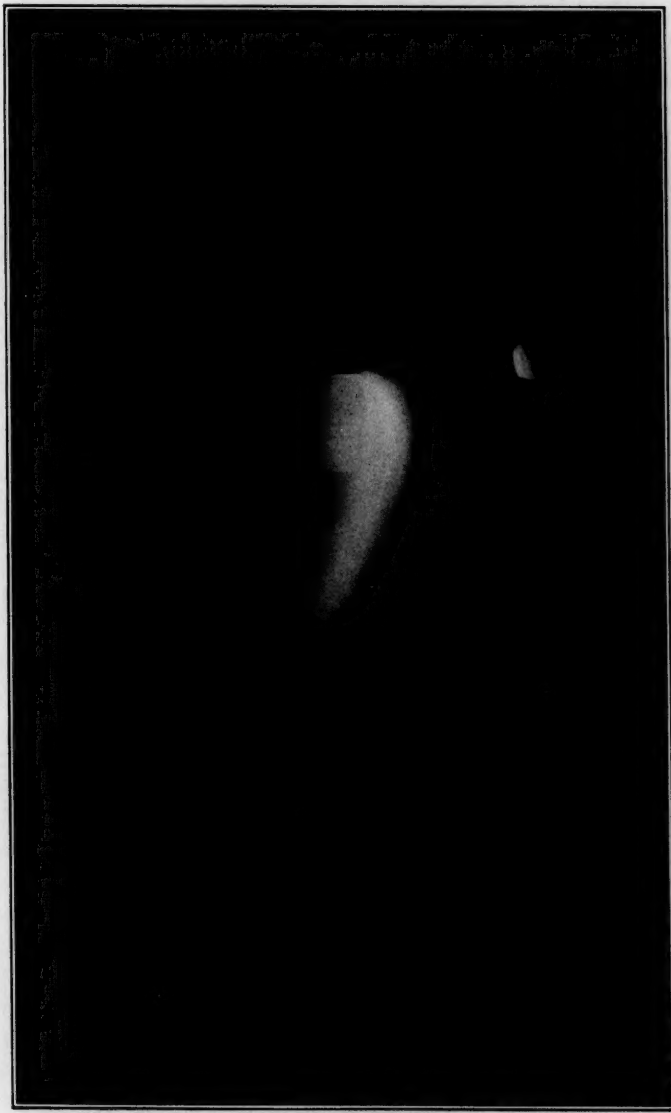
The Beck light submitted to the Navy was novel in the construction of the lamp and electrodes and the arc which was produced. The lamp was mechanically complicated and

needed the attention of designing engineers badly. It was designed to operate on 150 ampères, 75-volt arc, with rotating electrodes, automatic strike, and a constant mechanical feed on the negative, feeding it at a constant rate regardless of its rate of consumption. The positive was fed automatically by an unusual arrangement of a thermostat operating from the heat of the positive crater. The mechanical features of the lamp are sufficiently familiar at the present day and do not warrant detailed description here.

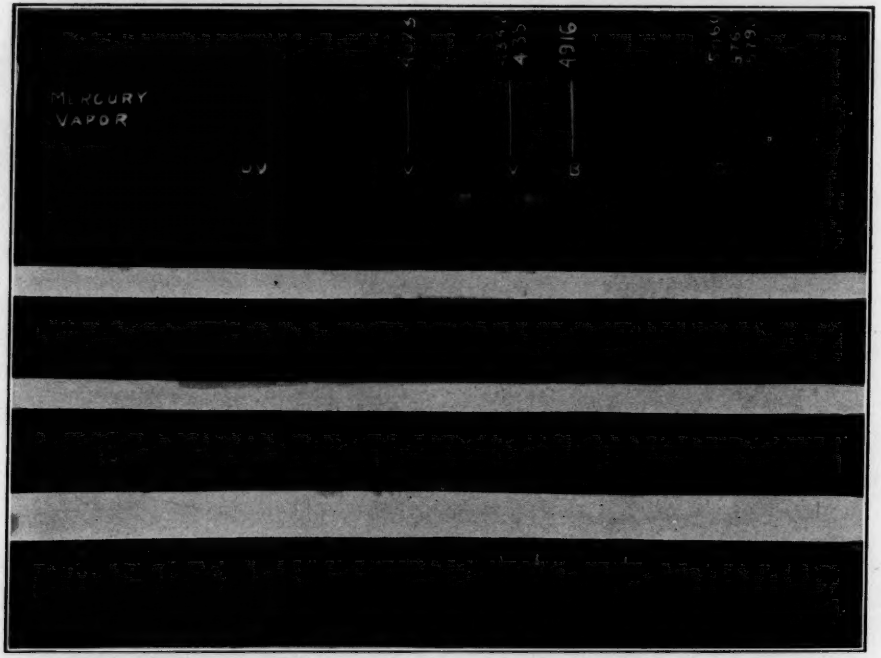
The really ingenious feature of the outfit was the electrodes. The old Naval lights of the 36-inch mirror size used 1¼-inch positive, while in this new light the positive was ⅝ inch in diameter and operated at 150 ampères, heretofore an unheard of feat. The negative was only 7/16 inch in diameter and instead of following the conventional trend of horizontal electrodes, the negative was inclined downward at about a 15-degree angle with the horizontal.

The arc produced by these electrodes was entirely new and investigation demonstrated that practically the entire useful light was emitted from an incandescent ball of gas held within a cup-like crater clearly and definitely formed at the tip of the positive electrode. This ball showed intensities as high as 80,000 candlepower, three hundred per cent greater than the old low power; and at the same time, as, it was only ⅝ inch in diameter in comparison with 1⅛ inch for the crater of the latter, it reduced the beam spread by about fifty per cent, when used under the same conditions.

Steps were at once taken by the Navy Department to adopt this light for ship's equipment. Mr. Beck supplied the original working model and the rapid development of this model by American manufacturers was remarkable. Various patents were issued for changes in mechanical and electrical design and methods of operating, to obtain high efficiency and greater reliability but the essential principle, as exemplified in the original Beck light of high current density and gas ball source,



THE BECK HIGH CURRENT DENSITY ARC AS FINALLY MODIFIED AND USED IN THE PRESENT NAVY STANDARD LIGHTS. THIS ARC BURNS IN AIR AND IS CONSIDERABLY STEADIER AND MORE EFFICIENT THAN THE ORIGINAL DESIGN OF ALCOHOL INSULATED ARC. THIS ARC IS OPERATING ON 13 K.W. BETWEEN THE ELECTRODES.



TYPICAL SPECTROGRAM OF THE BECK HIGH CURRENT DENSITY ARC OPERATING AT 80 AMPÈRES. EFFECT ON THE VISIBLE SPECTRUM PRODUCED BY THE GLASS OF THE PROJECTION SYSTEM IS SHOWN IN ADDITION WITH THE MERCURY VAPOR SPECTRUM AS A BASIS OF COMPARISON.

still remained as the fundamental feature of all modern high power arcs whether used in searchlights, in the motion picture field, or any other application.

The greatest difficulty from the first was found with the electrodes. These were something entirely new, both in size, composition and method of operation. The gas forming the characteristic ball in the crater cup of this arc, was found by analysis, to be not merely that of carbon but a mixture of the incandescent gases of various metallic salts carried by a hard core within the shell of the positive electrode and vaporized by the heat of the arc.

The familiar low power arc of eighty ampères or more, consisted of nothing more than the arc stream and flame, and illumination was obtained from the positive crater of incandescent carbon. The light from this source showed a complete band spectrum with a few definite bright lines, but in general the characteristics were those of a true band spectrum. The Beck arc with its incandescent gas ball gave radiations showing a distinct line spectrum. Analysis showed the presence in the core material of various metallic salts, predominant among which was Cerium. Spectrum characteristics, however, were complex. There was a continuous band spectrum supposedly produced by the incandescent carbon in the crater and superimposed on this a fine line spectrum of the rare earth elements in the core.

Incandescent brilliancy is ordinarily dependent on the temperature. Carbon alone, because of its high melting point, had previously been used to obtain these high temperatures. The temperature of the carbon arc is about 3600 degrees C. A more rapid volatilization of the carbon was produced by increase of current and no higher temperature obtained; nor, consequently, any greater brilliancy. Investigation thus showed that Beck was utilizing the incandescent gases of metallic salts in obtaining his high temperature and resultant brilliancy of light source, though still using carbon to carry the current and form the arc.

It was apparent Mr. Beck had found that, by using a carbon shell with an impregnated core of proper materials and forming a crater in this electrode, and then directing the negative arc stream of ionized gases into this crater, a luminescent gas ball could be produced and held in suspension in the crater, this ball having definite light giving characteristics of high intrinsic brilliancy.

Another great difference between the Beck and the ordinary arc was the high current density at which the electrodes were operated, the current density in the former being as high as 1.2 ampères per square millimeter of crater area. This high density produced the necessary heat to volatilize the impregnated core of the positive, and at the same time insured energy sufficient to produce a definite arc stream. The rapid volatilization of the core material caused the positive core to recede, forming a deep, cup shaped crater of parabolic formation and furnishing the recess wherein the volatilized gases were held.

The gas thus formed was held in place by the force of the arc stream from the cored negative which impinged on the positive gas ball, actually causing same to function under a slight pressure. The positive electrode was rotated to maintain the crater as a true cup with uniform sides to act as a proper and symmetrical container for the gas. In the original Beck lamp both the positive and negative electrodes were rotated. Subsequent American development failed to demonstrate the necessity for rotating the negative and this feature was abandoned, the negative in the American lamps of the latest type being fed mechanically but not rotated. The necessity for the rotation of the Beck negative electrode was apparently due to its long projection out of the head and to the desire to allow the complete bathing of this electrode by an alcohol flame.

In order to have the arc stream impinge on the ball in such a way as to hold it within the cup, it had been found necessary

to tip the rear end of the negative electrode downward thus permitting the arc stream to strike the gas ball at an upward angle of about fifteen degrees. The actual point of contact of the stream and the ball was determined to be slightly above the ball center, the axis of the stream being directed at the upper inside surface of the crater. This produced a rotation of the ball within the crater. Just why the revolving of the ball within the crater increased its efficiency was not clear, but such appeared to be the case. It is probable that it aided in the maintenance of its spherical form and retention within the cup. During the operation the excess gas from the ball flowed out of the crater causing the ball to project out in front of the crater in the form of a hemisphere.

Naturally the high temperatures caused rapid combustion of the electrode material, especially the shell, and thus the flame produced extended upward from the crater to about six inches. This flame had very little light giving properties in proportion to the crater and was not considered in figuring the arc intensity. It consisted of two distinct sections, viz., a high flame extending upward from the arc in a long tail, a characteristic of all arcs, and a short white flame of incandescent gas produced by the overflow from the crater gas ball. This last is a characteristic of high intensity arcs only. A discussion of this flame and its detailed characteristics will be later discussed in connection with the third electrode control.

Extensive research and investigation developed further interesting points in connection with this arc in its subsequent development. With its gas ball formed properly in the cup, the intensity appeared directly dependent on three factors,—current, arc voltage and solid area and shape of the crater. Of these the least important was the current, which was practically fixed by the positive electrode diameter and any appreciable increase of current over the normal caused smoking, while a surprisingly large decrease made very little change in arc intensity.

The arc voltage, however, directly controlled the velocity and definition of the arc stream and this was really the vital and controlling feature of the arc. The entire performance of the gas ball was dependent upon the direction, velocity and concentration of the arc stream. It might be well to explain that the concentration of arc stream refers to the fineness and definition of the stream, and is in turn, dependent upon the size of the negative tip from which this stream emanates. The negative electrode was also cored in order to aid in the arc stream concentration.

It has been found possible, with current density, electrode, and arc stream definition constant, to vary the intensity of the gas ball about twenty per cent above and below normal by simply varying the arc voltage. From this it can readily be seen that the adjustment of the arc to meet all conditions was rather delicate and such being the case, many difficult problems, dealing principally with holding uniform and correct crater formation, had to be met and to a large extent have been overcome at the present time. The greatest accuracy in the alignment of the electrodes relative to each other is essential and infinite care must be used today in maintaining this alignment, and the exactness of the negative angle. Exact gauges for this purpose are used in the assembly and furnished with outfits to ships, to check this alignment. Without it the highest efficiency in operation is impossible.

Spectrum analysis of this arc indicated characteristics of temperatures as high as 5500 degrees C. and optical pyrometers gave the same indications. Thermocouple pyrometers, however, did not indicate any such temperature, showing in fact only about 3800 degrees C. This would seem to indicate that some such effect is taking place in the gas ball as that which is found in the Welsbach mantle, where the material when heated to a certain point fluoresces, giving off radiations not produced directly by the heat applied.

Experiments concerning this feature have been conducted demonstrating that the spectrum characteristics of the gas ball can be changed by ionization of the gas beyond that point at which it operates. It may be possible that the superheated incandescent gas ball increases in brilliancy under the effect of the ionized negative stream actually fluorescing and giving spectrum indications of high temperature black body radiation. Research is needed to obtain more exact information concerning the true nature of this phenomenon.

The color of the light produced was blue white to the eye but the visibility curve of the light showed practically thirty-five per cent of the visible radiations in the green. The light was actinic, being better for photography than daylight and lending a means whereby night photography was possible for making pictures of large areas. An actual photograph was made of a tug located four thousand yards at sea and illuminated by a 36-inch 150-ampère searchlight of Naval type. This quality has rendered its use particularly applicable in the motion picture industry, for studio work. It is being used recently for projection purposes also.

Much credit must be given to American manufacturers for the rapidity with which the many difficulties connected with the production of satisfactory electrodes for use with this light were overcome. Manufacturers of the apparatus itself cooperated and the Navy Department, by establishing and efficiently equipping a laboratory for photometric work at the Navy Yard, New York, materially aided with both the electrodes and the arc mechanisms.

As at present manufactured, the electrodes of the Beck type are greatly improved both mechanically and from a photometric standpoint, over the originals supplied. The positive electrode is composed of two distinct sections, bound together to form a single, cored cylinder. The outside shell is constructed of carefully selected finely ground carbon. This shell is bored to one-half its diameter and within this pure carbon

shell is inserted a special core consisting of a combination of rare earth metals, carbon finely ground, and a binder. The secret of the manufacture of this core in its proper hardness, the proper proportions of ingredients, and proper specific resistance is held by the manufacturers, very little information being available outside of the trade. For successful performance it must first of all be chemically correct and it is further of about equal importance that it shall be mechanically perfect and sound.

The material, exclusive of the carbon used in shell and core, is obtained almost exclusively from the Welsbach Company, and it is understood that this "Welsbach residue," as it is called, furnishes the necessary rare earth constituents for the positive core. This rare earth material is obtained in the form of oxides but is reduced to fluorides and as such is used for core impregnation. The core when completed is extremely hard, and for this reason is termed the "Hard White Flame Core."

The shell of the positive is formed by squirting a pasty mixture through dies to its final form and the tube thus formed is allowed to dry. The core is formed by the same methods, after careful and uniform mixing of the ingredients. The core is then inserted into the shell, a binder being used to securely cement the core and shell. This must be done so as to absolutey fill all interstices between the shell and core, as empty spaces at this point mean useless electrodes.

The core and shell assembled, but "green"—that is dry but unbaked—are then placed in large ovens and baked until the whole mass is one hard solid electrode. The baking process is very slow requiring about twenty days during which time an even temperature must be maintained. The "green" electrode must be packed in sand within the oven and this packing must be very carefully done so as to present an equal distribution of the weight of the entire mass. In the "green" state the electrodes are more or less flexible and have a tendency to

warp or bend and this tendency is overcome by this method of packing. Further, changes of temperature, or the uneven application of heat to the oven as a whole, will also cause warping, and damage to the electrodes which will render them unserviceable. Due to the extreme length of the electrodes, the baking process is rendered very difficult and only by the exercise of exteme care, will good electrodes be produced. This baking is probably the most difficult single operation in the manufacture of electrodes.

The present electrodes while exhibiting many characteristics which are similar to the old white flame core of the well-known flaming arcs, are not the same in all respects. The old white flame core, now distinguished as the "soft white flame core," will not operate as a high current density electrode. The use of the soft core destroys the vital crater formation, thus distorting the gas ball and after a short period of operation fails to either produce or hold any of the desired characteristics of the Beck arc. The "hard white flame core" electrode had its advent with the Beck arc.

The negative electrode is also cored, similar in mechanical construction to the positive, but this electrode is made entirely of carbon. The shell is manufactured in about the same way as the positive, by being squirted through dies, but the core is not formed prior to insertion in the shell. The shell is baked without the core and the core material is then squirted into the shell and allowed to dry without baking. The negative shell is not as hard as the positive and has a much coarser grain. The core is what is known as a "hard pin core," of very much finer and closer structure than the shell. This construction of the negative allows it to burn to a long tapered point with the core, harder than the shell and consequently burning at a slower rate, protruding in the form of a pin about a sixteenth of an inch beyond the shell. This combination of slow-burning core and rapidly-burning shell, prevents the condition, experienced on the first electrodes developed, where

the core burned back within the shell during operation and left a crater at the end of the negative tip. Unless the core retains proper relation to the shell while burning, a wandering arc and a poorly concentrated arc stream will result. The material of the negative in no way contributes directly to either the color or the intensity of the light source.

The physical and mechanical properties of the positive core and shell must be such as to produce a full gas ball and hold a constant and well-defined crater during operation. A peculiar feature of the cored electrode is the relative electrical conductivity of the core and shell. The core and shell, being made separately, are practically two separate units when burning. When cold, the shell conductivity is higher than the core. This makes possible the admission of the current to the positive electrode by means of surface brushes, with no other means of distribution of current throughout the electrode. As the electrode is heated the core gradually increases conductivity at a rate higher than the shell, until at the hot crater the greater percentage of the current is transferred to the core. The electrode, being cold under the contact brushes and reaching its maximum temperature at the crater, causes a gradual reduction of the shell current and an increase of the core current from brush to crater. The shell and core form two parallel paths in electrical and mechanical contact, reversing their relative conductivity gradually along the length between the point of admission of the current and its exit at the crater.

Practically the entire current, therefore, at the crater passes through the core, utilizing only one quarter of the total electrode, and producing intense heat which volatilizes the core rapidly into gaseous form. The shell in the meantime which is in contact with this hot gas, is consumed, its rate of consumption, by virtue of its construction, being proportional by length to the core consumption and is entirely controlled by the rate of the latter. These two parts of the positive elec-

trode, core and shell, are so nicely proportioned that when the core has receded a given amount (actually half the shell diameter) the confined gas ball radiates sufficient heat to consume the shell at the same rate as the core, thus maintaining a constant definite crater depth.

A further effect, produced by the passage of all of the current through the core at the crater, is that by this means all current must be passed directly through the gas ball, thus highly superheating the gas composing it. The arc, too, does not wander over the positive as in the case of the old carbon arc, and the arc stream retains its definite direction and point of impact on the gas ball, aided by the coring of the negative.

The advantage of the impregnated core alone for any arc was early seen by the Navy Department and the old style electrodes were requisitioned cored with white flaming material even at the time the high power electrode was being developed mechanically in this country. The use of the white flame core in the old electrode produced no additional intensity, but it actually increased the effective range of a 120-ampère old style lamp from about 2500 yards with the old neutral cored electrode to 3500 yards with the white flamed cored type. This in itself was a great step forward exclusive of the arc. The impregnating of the cores of the low power electrodes with white flame material, similar to the hard white flame core of the high power type, did not by any means produce a high power gas arc on the old electrodes, but the addition of the core impregnations produced a colored gas in the crater which changed the beam characteristics.

These high power electrodes, due to the delicacy of the task which they must perform, as previously indicated, must of necessity be as nearly mechanically perfect as possible. Shell splits, bubbles extending into the shell, bubbles or blank spaces in the core, and loose cores, all result in electrodes that will not function properly.

The greatest difficulty which has confronted manufacturers to date, is the loose core problem. It has been definitely determined that a loose core will not give efficient results, the effect varying from a slight loss in intensity to total failure to function on the part of the electrode. This fact and the fact that there is no known method of finding exactly how loose a core is, beyond the mere fact that it is loose, has cost the Government many thousands of dollars in rejections. The rejections a short time ago were running as high as 40 per cent of the contents of an oven for certain manufacturers, while others had succeeded in perfecting their methods whereby the rejections were cut to as low as 15 per cent. Rejections, while not apparently a direct charge, are paid for eventually by the consumer.

A core may be slightly loose in spots and cause unsteady burning or, at the worst, a loss of crater until the bad spot is consumed. This consumption may take from one to ten minutes dependent on the extent of the loose area. This at first thought does not seem a disadvantage warranting any extensive rejection of material. But when it is considered that the lights are to be used for military purposes where a minute lost may be of vital importance, the case presents a different aspect.

As a very striking example of what the "out" of a light may mean, an American destroyer just prior to the war, was leaving a Mexican port, the entrance to which was protected by sea walls, with a fairly narrow, navigable channel. The destroyer's exit was made after dark and in order to safely maneuver, the forward searchlight was put into operation. At a critical moment the positive electrode "blew" its core, that is the core was not continuous within the shell and the expansion of the gases within the black space, caused the forward end of the core section, about an inch and a half in length, to blow out of the shell, leaving a hole one and a half inches deep. The arc was immediately extinguished and failed

to strike again. The destroyer was making good headway and before the after light could be started and trained properly, had missed ramming the sea wall by a dangerously narrow margin. This illustrates what may happen and how reliable in operation, electrodes and arc apparatus must be to meet ordinary navigational requirements. Electrodes must be carefully made and as nearly mechanically perfect as possible to be at all trustworthy.

Many questions relative to electrodes for the high power arc still remain to be answered. And it is in this direction that, it is believed, the greatest possibilities for improvement exist. Once research produces improved electrodes, the electrical and mechanical features of the present lamps can be quickly adapted to utilize them.

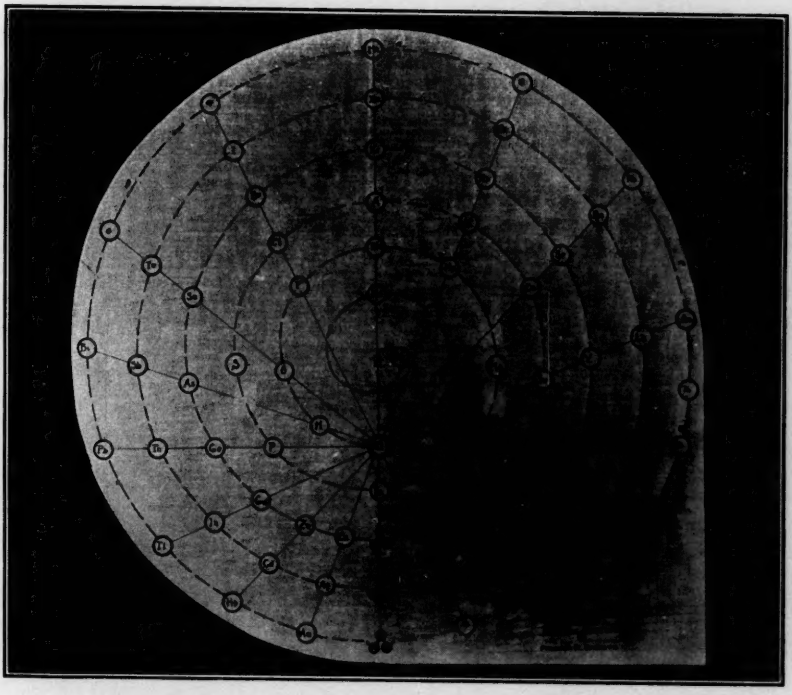
Peculiar conditions of service under which the present lamps are used have resulted in failure to realize practically all of the illumination of which theoretically they should be capable. All are familiar with the fact that on certain nights long range and excellent visibility are obtained, while on others, with apparently identical atmospheric conditions, the results are most disappointing. Again observers in the immediate vicinity of the lights will fail entirely to see the object illuminated while others further removed see it clearly. A vessel of war operating such a light becomes an excellent target to her opponent. If she cannot quickly obtain sufficient visibility for an effective attack the lights may mean her destruction.

Investigation and experiment have developed methods of operating with these lights which have eliminated some of the difficulties, or have made it possible to operate effectively in spite of them. However, it has been shown that the distribution of the light of the arc in the visible spectrum is not all that could be desired and that an improvement in this respect is greatly needed.

Spectrophotometric analysis of the beam of the high power arc indicates that there is at present far too much of the

violet, blue and blue-green light included. By experiment, under certain atmospheric conditions the violet, the blue and some of the green were reflected by the atmosphere and practically removed from the beam at a distance of five hundred yards from the arc. A beam, which showed high intensities in the visibility curve in the green, gave very low intensities at this point and in fact the entire shape of the visibility curve underwent a selective change within five hundred yards. It has been further determined that this light, which had been removed, was not absorbed as might be expected, but was reflected and refracted by the atmosphere, causing a fluorescent haze around the light, even showing fairly high intensities back of the drum. This condition was more or less pronounced under atmospheric conditions where moisture, similar to dew, filled the air. Under this same condition the beam at five thousand yards was strictly orange in appearance and the intensity on the visibility curve in the orange was high, but due to the conditions at the light, the operators located within two hundred feet of the same, could not see the objective. This seemed to explain the failure, sometimes experienced by operators in the vicinity of the arc, to see the objective. The reflected light, extracted from the beam by the atmosphere, formed a luminous blanket completely enveloping the light and its vicinity through which visibility from near the arc is difficult or impossible. The retina of the eye adapts itself to the intensity of this blanket which is, in general, higher in intensity than the illumination at the objective and as a result, to observers within the blanket, the objective is for practical purposes invisible.

Following these findings and with a view to remedying the effect caused by the reflection of certain portions of the beam light, efforts have been made to change the beam color. It was hoped to progress along this line by the production of a gas ball having a yellow beam of spectrum characteristics showing high intensities in the yellow orange. To accomplish



ARRANGEMENT OF THE ELEMENTS IN THEIR PERIODIC SYSTEM.

this required an extensive study of the various materials with which it is possible to impregnate a core of a high power electrode and which might produce the desired results.

During the original development of the high power carbon Mr. Preston R. Bassett, then employed on research work by a manufacturer interested in the development of high power electrodes, observed certain characteristic phenomena pertaining to the materials with which the core was impregnated. He developed a schematic arrangement of the elements in their periodic system and according to their atomic weight and other characteristics from which it is possible to group and identify the materials which will operate in high power electrodes as core impregnations. In this arrangement it is found that elements having similar properties fall on certain parts of his chart in the immediate vicinity of one another, and are easily identified. If one material is used and its results known, by consulting the chart, several other materials which will produce the same result can easily be picked out. A reproduction of this chart is given showing the layout of the various elements and the group which it has been found possible to apply to the core material. This chart is reproduced by permission from Mr. Bassett, whose efforts materially aided the original development of the American made electrode.

Chemical analysis of the original Beck Electrodes showed a predomination of cerium. Consulting this chart we find that any of the materials falling within the same zone should give results similar to those obtained from cerium. Actual experiments proved this to be a fact, but further showed that the cerium was the best fitted for core impregnation, due to other governing conditions, such as the readiness with which it could be procured on the commercial market, and its performance in the crater when operating under high temperatures.

The original Beck electrode further used calcium in the core and its function was thought to be for the purpose of coloring the beam to produce substantially a white light, similar in ap-

pearance to the light produced by the old calcium spot lights. This led to the belief that other elements falling in the same zone as calcium would be useful in changing the beam color, if substituted for calcium. The two elements adjacent to calcium were tried, namely, strontium and barium. Strontium was tried first, it being known that the characteristic of a strontium impregnated arc is red, but efforts were without result. Barium was next tried in an attempt to color the beam a characteristic barium green, but still without result.

In the efforts to develop along this line the limit of the experimental possibilities was rapidly reached, and it was eventually demonstrated that the material with which the core was impregnated had no apparent bearing on the beam color. The governing features of the core material appeared to be only the production of steady burning and the ability to hold the physical balance of the arc. The material with which the core was impregnated certainly changed the flame color and gave a beautiful colored tail beam and foreground light, but the projected parallel beam retained its original characteristics under various changes of the metallic salts in the core and the color of the gas ball was apparently in no way altered.

The fact that the beam is produced by light emitted from the gas ball and that the color of this ball is practically independent of the core material, renders an attempt to change the color of the beam an exceedingly difficult matter and its solution is still to be found. Experiments have shown that any material which will maintain the characteristics of the Beck arc will produce a beam of the characteristic blue-white color with similar visibility and energy distribution in the visible spectrum as in the original Beck lamp.

Thus a logical attempt to solve what at first seemed a simple problem, proved a failure when practically applied. Further the use of the color screens to solve the difficulty could not be considered for various reasons, the main one being their high absorption of energy throughout the entire spectrum. Gold

mirrors for a time seemed to answer the question and proved very satisfactory with the low power arc but when used with the high intensity arc, they proved so inefficient as light projectors that the idea was abandoned. Experimenting along this line is still under way, and it is hoped that some day a beam of suitable color will be produced without the loss of any of the present high illuminating qualities.

While the problem of suitable electrodes was and still is, perhaps, the most serious confronting those engaged in the original development of the high powered arc, there still remained others, of almost equal importance, concerned with the satisfactory production of the arc from these electrodes and its maintenance and the suitable projection of the illumination. For example, the original Beck lamp used an alcohol flame surrounding the arc. The exact reason for this use was at first somewhat of a mystery. It subsequently developed that the purpose of the alcohol was to furnish, in effect, an air insulator for both the positive and negative electrodes within the area of high oxidizing temperatures produced by the heat of the arc. The entire arc and exposed portions of both electrodes were enveloped in an inert gas formed by the combustion of the alcohol, and what is known as "air burning," or consumption of the electrodes other than at the arc itself, was prevented.

A demand for the use of these lights in anti-aircraft defence, which developed almost simultaneously with their introduction into this country, resulted in the condemnation of the alcohol for this purpose almost at once, as, when it was used, rotation of the drum in the vertical plane was not permissible.

The first successful method employed to eliminate the use of alcohol was the introduction of a quartz sleeve extending as a projection of the positive nose and covering the positive electrode to within five-eighths of an inch of the crater. Quartz is highly refractory and easily obtainable and air insulation was accomplished by the fact that the inert carbon dioxide

gas, produced by the hot carbon in contact with air, filled the air space between the quartz bushing and the electrode, and protected the latter back to the contact brushes. Beyond the brushes the temperature was not high enough to result in appreciable air burning. The negative was operated without protection as the negative could be burned close to the metal of the negative head without danger to the latter, and experience proved further protection to be unnecessary. This is mainly due to the fact that the greatest amount of heat is released at the positive crater and also to the characteristic behavior of all arcs, in that an established arc will not transfer itself from one negative to another irrespective of what the relative difference of potential may be. There is thus no danger of the transfer of the arc from the negative electrode to the metal of the negative head with consequent destruction of the latter. Such is not the case with regard to the positive, as an arc will readily jump from one positive to another or operate on two or more at the same time.

At the same time the introduction of the quartz bushing made it necessary to artificially cool the positive head in order to keep the temperature at a minimum for long sustained periods of operation. This feature was accomplished by building the positive nose hollow to form an air chamber which permitted a circulation of air through the head, care being taken to keep possible drafts away from the arc.

The original Beck and the first production of American manufacturers did not use any artificial head ventilation. They were designed with a great volume of metal in the heads. This metal acted as a radiator and cooled the parts of the head to a workable temperature. Thus the Beck and first American design were exceedingly head heavy, making them difficult to construct for universal application as anti-aircraft lights. The heavy heads suspended on their long supporting columns, would droop when high drum elevation was attempted, throwing the arc out of focus and upsetting the arc control system.

By artificially cooling the heads, their weight was reduced to about one quarter of that of the first type. This was apparently the only course for the added reason that the columns had to be so constructed as to obstruct as little light of the beam as possible.

However, though a material advance, the lights using the quartz bushings soon proved unreliable in service. During long periods of operation or as the result of inattention on the part of the operator, conditions developed which frequently put the light entirely out of commission. The entire positive head exclusive of the quartz was of positive polarity and a wandering arc which often resulted in service from many causes, was followed invariably by a transfer of the arc to the metal of the head, back of the quartz, and in a few seconds the entire front mechanism would be a mass of molten metal.

The negative electrode at this time, too, was copper coated throughout its entire length as a further protection against air burning. In the elevated position for anti-aircraft defence this resulted frequently in the deposit of globules of molten copper on the mirror face, pitting and weakening it so that mirror breakage from this cause was not unusual.

In spite of their unreliability, however, development for at least the thirty-six and thirty inch sizes remained at a standstill from this point to the end of the War. They gave a wonderful beam, when they operated, but their continuous operating period was short and depended greatly on the skill of the operator.

In the twenty-four-inch size, a non-automatic type using the high intensity principle was the first production, followed shortly by an automatic type much the same as the larger sizes but in place of the quartz bushings the metal of the positive nose was extended to within three quarters of an inch of the crater. This type being designed for constant service mainly in signaling, was very rugged and satisfactory in mechanical design.

Metal was used, however, close to the positive crater. Special design was needed to radiate the heat and this combined with lower temperature at the crater, reduced the danger of the transfer of the arc to the positive head except in anti-aircraft position where danger to the positive head still was excessive. Even in normal horizontal position the burning of the positive head was an ever present possibility, through some unusual condition, such as a cracked crater due to hair split in an electrode, feed failure for a short period, or lack of proper thermostat adjustment. This type was, on the whole, fairly satisfactory.

Early in 1917 a new type appeared in the twenty-four inch 80 ampère arc, size. This type incorporated in its design an entirely novel means of positive control, eliminating the thermostat. This feature was the third electrode positive control. Subsequently facts developed from investigation of this feature which aided in eliminating danger of head burning in the larger sizes, though for its original purpose it has never proved satisfactory for lights of greater current densities than the twenty-four inch lamp.

The action of the third electrode is believed worthy of as detailed a description as possible. It has been and still is the subject of considerable controversy.

The flame of the high intensity arc has been previously described as consisting of two distinct parts. The small flame when analyzed is found to run to temperatures only slightly less than that of the crater itself and its resistance per millimeter is low. The larger flame has a relatively low temperature and its resistance is very high, making it relatively an insulator in comparison with the small flame. This small flame is termed the "control flame" in the high power arc lamp.

These two flames as they extend above the crater are out of the mirror focus, and therefore, while they add perceptibly to the total arc, in no way enter usefully into the beam. The large flame is a very poor light producer but the small control

flame is practically equivalent in light production to the crater itself, being highly incandescent. This flame projection extending only about one quarter inch above the crater tips, is the cause of the beam tail light. While it has disadvantages which make it rather objectionable for both Naval and Military lights, it has been put to work to simplify a necessarily complicated mechanism.

The control flame at all times holds a definite relation with respect to the crater tip, the flame moving with the crater as it is alternately consumed and fed. For this reason and in view of its particular electrical characteristics, it lends itself very readily for use in the third electrode system. This flame is actually an incandescent and highly ionized stream of volitalized electrode material with definite conductivity and uniform characteristics.

The operating control by this method is identical with that of the thermostat so far as the mechanical and electrical features are concerned. The control rod is operated by an electromagnet. The positive terminal of the electrical circuit of the control magnet is connected to the positive side of the arc, the current passing through the coil, and the negative being the third electrode.

The third electrode itself is mounted on the positive head nose, directly over the positive crater tip. The operation is accomplished as the "control flame," moving backward with the positive electrode, when same is consumed, comes in contact with the third electrode. The stability of the flame is such that its contact with the electrode can be adjusted to produce any desired voltage, the voltage depending upon the area of the flame in contact with the electrode. By virtue of this fact, the fineness of the feed and the sensitivity of the circuit can be nicely adjusted.

The magnet is designed to produce a certain flux at a set current and at a predetermined voltage. As the flame comes slowly back to the electrode and finally makes contact, voltage

is developed and current passed. As the flame continues to feed further back, it bathes the electrode to a greater extent, increasing the voltage by virtue of its resistance being inversely proportional to its area of contact, until the current in the circuit produces the designed flux to operate the positive feed.

The positive electrode is then fed forward in small steps. This gradually moves the flame forward away from the third electrode. The flame will pass current as long as it is in contact and sufficient residual magnetism remains in the magnet to hold the feed until the flame is actually out of contact with the third electrode. A spring return of the magnet armature is provided, which opens the magnetic circuit when the current has passed below a certain value. As in all magnets, the armature air gap acts as a dash pot for the closing point, and once closed very little flux will hold it closed. This phenomenon is very useful in the operation of the control as it avoids the necessity for holding the flame in actual contact with the third electrode, and thus prevents it from being consumed. During the operation of a lamp the control flame is actually in contact for about twenty per cent of the time.

This third electrode must, of course, be insulated from the positive electrode and, if it is firmly fixed in position, it will insure the holding of the crater within three millimeters of a predetermined point with respect to the positive nose.

This has a particular advantage in that by mounting the control system of the positive feeding mechanism on the lamp, the entire control of the electrode feeding is removed from any connection with the drum. With the thermostat control as supplied, up to the time this light appeared, the thermostat was, on both twenty-four and thirty-six inch types, always mounted on the drum. This meant a definite and fixed relationship between the setting of the lamp and the drum.

The thermostat control up to the time of the third electrode development permitted only one definite lamp setting and if this setting was incorrect, the efficiency of the beam was disturbed

and the defect difficult to remedy. The third electrode control acted only to keep the positive crater at a predetermined distance outside the positive nose, independent of the mirror focus, thus permitting a focus of the beam, to any desired spread, at the will of the operator. In order to change focus it was necessary to change the relation of the lamp to the drum, hence the lack of flexibility of the thermostat mounted on the drum. This led eventually to a change of the position of the thermostats on the latest of the larger sizes from the drum to the lamp itself.

The electrical circuit of the third electrode is merely a high resistance shunt to the main arc circuit. The third electrode is negative to the magnet circuit but positive to the arc and current is thus passing into the arc from two separate positive sources, which phenomenon is theoretically feasible. It is merely a case of the current carrying path being established by the highly ionized gases of the control flame and the amount of flow of current is directly in proportion to the inverse ratio of the resistance of the two circuits.

At first thought it may seem doubtful how the positive main circuit could contain sufficient resistance to allow an appreciable voltage and current in a parallel circuit of apparently high resistance. This point may be cleared up by an analysis of the arc voltage drops from the positive to the negative electrode. There are four zones of resistance existing in the arc. They are the resistance of the gas ball itself, the contact resistance of the gas ball and electrode, the resistance of the arc stream between the gas ball and the negative, and the contact resistance of the arc stream and the negative tip.

The value of these resistances, measured in terms of voltage drop, is considerable. From positive electrode tip to negative tip there is a total drop of sixty-eight volts in the 80 ampère arc. This is made up of a sixteen volt drop due to positive tip resistance R1, four volts in the gas ball R2, thirty-five volts

in the arc stream R3, and eighteen volts at the negative tip R4. These values may vary slightly but they are a fair average.

The third electrode circuit is shown in the diagram, Fig. 1. It consists of a shunt around both the electrode tip and the gas ball, and a drop of twenty volts is thus possible. As has been stated before, the control flame has a definite resistance per unit of length, so for fine operation this electrode is placed as close as possible to the positive tip, thus making the control flame resistance as small as possible. The control flame resistance is figured in that of the control circuit. This control circuit operates on .025 ampères, at seventeen volts. During operation, due to the fact that the large tail flame has a very low but still an appreciable conductivity, current is always flowing in this circuit, although naturally a very small quantity when the control flame is out of contact. For this reason the operating magnet requires a spring return.

Many theories on this system have been advanced, among which is the theory that the contact of the ionized control flame with the electrode block actually generates an E.M.F. of its own. The magnitude of this voltage is supposed to be proportional to the extent of the contact surface between the flame and the third electrode, the electrode thus being positive to its own circuit and the flow of current opposite to that described. Experiment along this line has failed to prove any such fact. It is actually possible to obtain an auxiliary current with the third electrode, over either the positive or the negative arc electrode, obtaining a maximum drop of twenty volts at the positive and about thirty-six to thirty-eight at the negative. When the control is operating at either the positive or negative, however, the circuit connection must in either case be made from the positive electrode, thus actually resulting in two positive paths to a single negative. Even two separate shunt third electrode paths, one operating from positive to control flame, and one from positive to negative arc stream fringe,

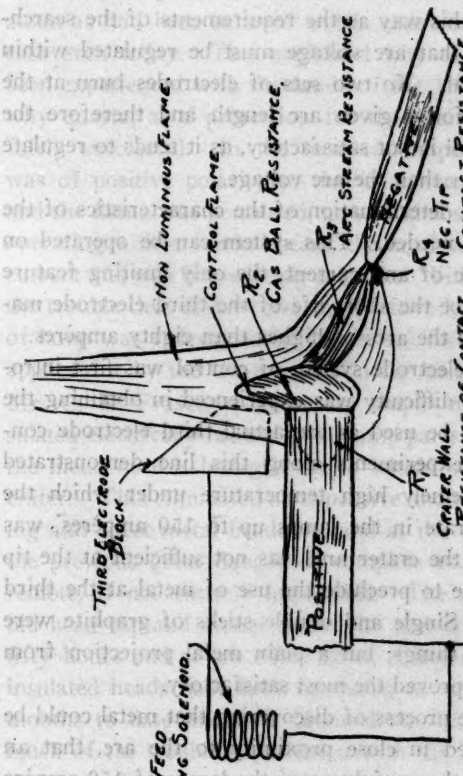


DIAGRAM OF ARC RESISTANCES & THIRD ELECTRODE CONTROL CIRCUIT.
FIG. 1

are possible. This identical arrangement is being used in the latest type of motion picture projector automatic high power lamps, giving a simple control for both positive and negative. The negative feed control on Naval searchlights has never been attempted in this way as the requirements of the searchlight arc are such that arc voltage must be regulated within two volts of normal. No two sets of electrodes burn at the same arc voltage for a given arc length and therefore the negative feed control is not satisfactory, as it tends to regulate the arc length rather than the arc voltage.

A more complete determination of the characteristics of the third electrode are needed. This system can be operated on any high power arc of any current, the only limiting feature to date appears to be the short life of the third electrode material when used on the arcs of higher than eighty ampères.

When the third electrode system of control was first introduced, considerable difficulty was experienced in obtaining the proper materials to be used as the actual third electrode contact. Subsequent experiments along this line demonstrated that even the extremely high temperature under which the electrode must operate in the lamps up to 150 ampères, was confined mainly to the crater and was not sufficient at the tip of the control flame to preclude the use of metal at the third electrode contact. Single and double sticks of graphite were tried among other things, but a plain metal projection from the obdurator plate proved the most satisfactory.

It was during the process of discovering that metal could be successfully operated in close proximity to the arc, that an advance increasing the usefulness of the lamps of 150-ampère size, both in efficiency and reliability, was accomplished, as previously mentioned.

It was proved that though a metal block would operate satisfactorily on the minute currents required for a control circuit, yet, should the resistance of this circuit be reduced to allow the transfer of an appreciable current by the control flame

to the electrode block, the block would immediately melt. This plainly indicated that the frequent burning of positive heads on other types had been mainly the result of the current carried by the metal of the heads and not the heat of the arc alone as had been believed and is claimed by some to be the case even today. This explained why so much trouble had been experienced with the heads of the 150-ampère lamp, and why quartz, as a protection for the positive nose, had been fairly successful. In the first 150-ampère lamps the head was of positive polarity so that if contact between the head and the flame was permitted, as was almost unavoidable when operating with a vertical beam, the head metal began to carry current and was immediately melted. With the true cause for the trouble thus understood, the positive head cap of one type of lamp was redesigned by the Naval Laboratory and patent applied for by the Government in the name of its originator, Mr. C. E. Fraser, a laboratory employee. This cap is insulated electrically neutral to all lamp parts and electrodes, and the metal nose is brought out to within one half inch of the crater. This eliminated an expensive and delicate quartz bushing and sheet metal head cap, and at the same time rendered the previous great danger of complete head destruction very remote, if not entirely impossible. The average life of the old head cap and quartz under the best conditions was about fifty hours and at sea often about thirty minutes. The new insulated heads have been operating in service for about ten months to date and it is believed that they have eliminated much of the trouble experienced with the lamps using the old quartz bushings and insulated sheet metal head caps. The insulation consisting mainly of mica discs must be frequently inspected and kept intact. If it breaks down the head will be in danger of destruction as formerly.

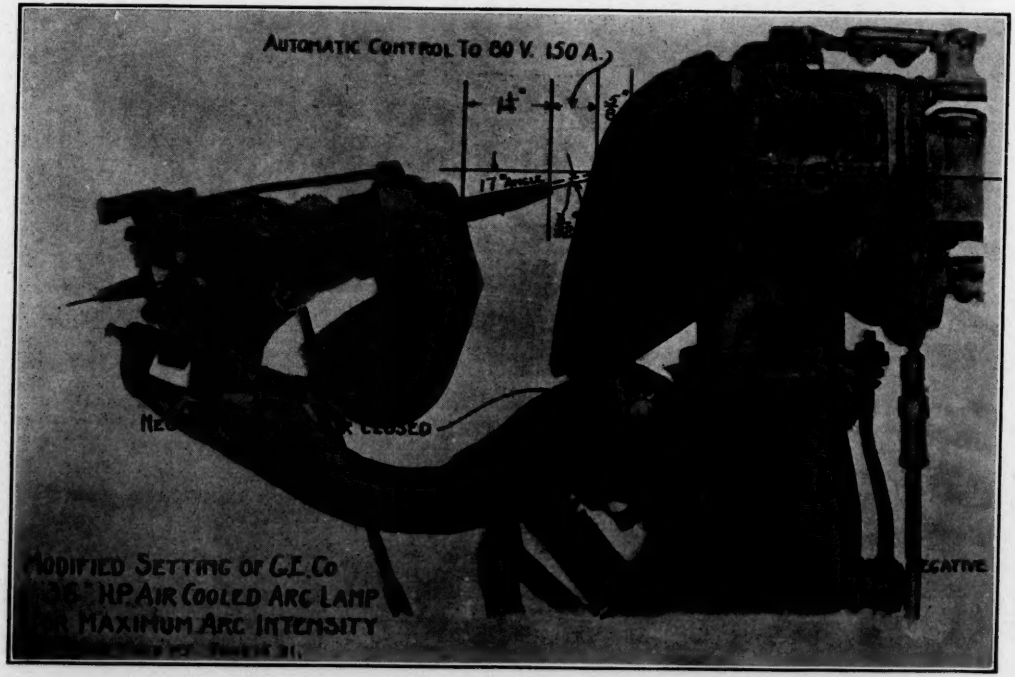
This inspection and testing of the insulation is an easy matter. It is merely necessary to ring with a battery and bell or magneto between the front of the nose and the head

proper. The importance of this insulation of the head cannot be overestimated and should be tested periodically and before and after long runs. Faulty insulation may lead to damage to other vital parts of the lamp including mirror breakage. Immediately this cap and nose carry current even in small amounts, melting of the metal will start at the point of contact of the current leakage and this condition will rapidly magnify itself until the entire arc will transfer to the head, causing its destruction.

The delay in discovering this feature of insulating the positive head nose was due to a peculiar condition in the development of the lamp, where everyone concerned took for granted the fact that with temperatures as high as those of the crater, metal must be kept away from the arc. This idea apparently was held by Beck, and in the development of the American lamp about every other scheme but the insulating of the head was tried. Like many developments, the hardest means seems to have been tried first and the simple discovered only by hard experience.

During experiments with the new head it was further found that greater efficiency of illumination and steadiness of burning could be accomplished by permitting the negative to air burn back for about one and a half inches, thus forming a sharply pointed negative tip giving great definition of arc stream and high stream velocity. This was accomplished by cutting back the negative nose, permitting the negative to project one and a half inches outside the head, and by removing the copper coating of the negative, exposing the entire hot end of the electrode directly to the air. This is a radical change from both the original Beck design and the subsequent American development. Both of these prevented air burning of the negative by protecting the electrode to its tip. It naturally reduced the life of a trim but not to a prohibitive extent.

The third electrode control gave a rugged design and eliminated the difficulty with thermostats on arcs of less than



VIEW OF HEADS SHOWING THE EXACT ALIGNMENT OF THE ELECTRODES DURING OPERATION TO PRODUCE THE MAXIMUM ARC EFFICIENCY.

eighty ampères. The radiant heat emitted from an 80-ampère arc, while it could be concentrated to operate a thermostat, gave a factor of safety for positive operation far too small and tended to introduce irregularities of feed. The third electrode, however, was found not suitable for the larger sizes where perfect arc balance is a more vital factor. It is sometimes used on larger sizes, however, but only as an auxiliary to the thermostatic control, supplying a safety device should the latter fail.

One vital feature in the building and operating of lamps for the high power arc was found to be in the mechanical alignment and relation of the positive and negative electrodes. Experiments showed that a critical angle existed between the positive and negative electrodes and further, that the center lines of the two electrodes must be in the same horizontal plane. Unless these definite relations were strictly adhered to, unsteady arc, poorly formed gas ball, and result and loss of efficiency were sure to be experienced. It was found that a tolerance of not more than .0025 either side of the positive center line could be allowed for negative alignment and that the negative angle must be adjusted to within one quarter degree of the theoretical seventeen-degree angle. This close alignment must prevail at all times during operation and the mechanical construction of the heads must be such as to hold this alignment permanently. It was further found that, in setting, the relative positions of the two electrodes in their proper planes, controlled to a great extent the intensity of the arc and the steadiness of the gas ball within the crater cup.

In order to make this delicate alignment during the manufacturing process, it was found necessary for the manufacturers to build delicate and expensive special jigs and gauges for this work, and on this point most rigid inspection is conducted by the Navy Department to insure a perfect setting.

Another vital factor in the development of the present day lamps was the drum ventilation. Without ventilation heavy

white deposits formed on mirror and door and after fifteen minutes of operation these deposits were so dense that the usefulness of the lamp was destroyed. Further the excessive heat of the arc and the high heat absorption of the mirror caused extreme temperature variation in the latter and consequent probability of breakage. The actual high temperature of the mirror is not necessarily conducive to mirror breakage, but it is absolutely essential that the temperature over the entire surface of the mirror be kept uniform and free from zones showing great difference in temperatures. High grade mirrors were necessary to meet this danger and an improvement of the drum ventilation system of the Beck light which though supplied, was extremely inefficient.

Two general systems of ventilation have been worked out by American manufacturers and are now in operation on our lights in service. One is known as the plenum system, or what is in effect a pressure type, and the other is the exhaust system.

In the first or plenum system, the air is forced into the drum under a definite pressure above atmospheric and at high velocity. A blower draws the air directly from outside the drum, discharges it through ducts over the back of the mirror, thence to other ducts opening longitudinally along the inside of the barrel, and finally forces it out through a hood at the top of the drum, carrying soot and gases with it. Some disturbance of the arc is unavoidable by the eddying currents of air inside the drum. This is an objectionable feature as is also the probable variation of temperature between certain zones of the mirror, due to the arrangement of the ducts, and the unavoidable pocketing of air, thus increasing possibility of mirror breakage from uneven cooling. This type uses only one motor for both lamp mechanism and ventilation. This is an advantage.

In the second, or exhaust system, the blower is placed in the discharge hood on top of the drum and sucks air from the

drum itself, discharging it directly to the atmosphere. The pressure in the drum is thus slightly below atmospheric and the velocity of air within the drum and the ducts is low, and the ventilating air blows gently in through both drums and mirror ducts in parallel. It is at once apparent that this system causes less disturbance of the arc. Both are efficient but the second system is favored for the reasons given and because it is believed to result in more uniform, though higher, temperature over the area of the mirror. The subject is still being investigated at the present time. This type uses a separate motor for ventilation alone.

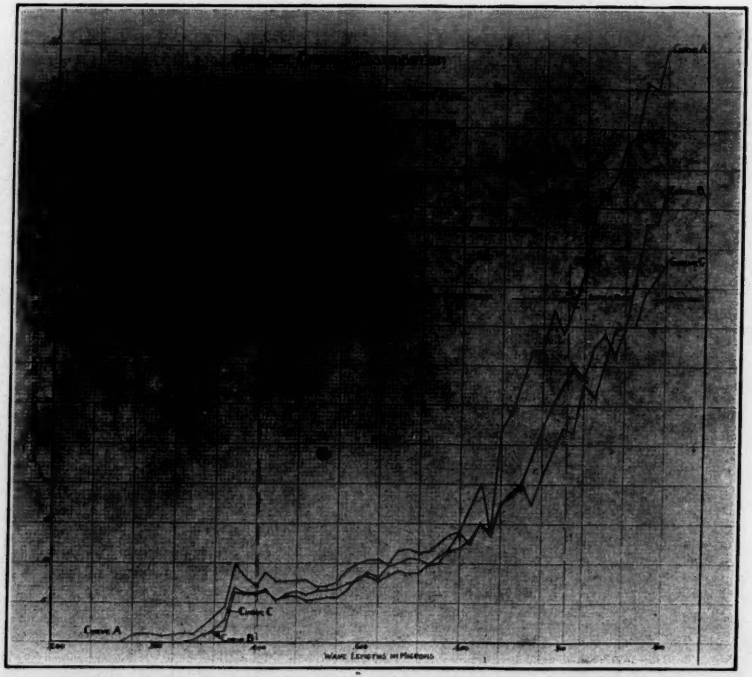
Two American manufacturers in developing this light differ also rather widely in another vital feature. Both have obtained results and both types are exemplified in our present service lights. This feature arises from the necessity for a rather accurate control of both voltage and current. One system operates the negative to keep the arc current at a set value regardless of arc voltage, line voltage or arc length. The second system operates the negative to hold a set value of arc voltage regardless of arc current or arc length. The two are radically different, mechanically and electrically.

Discussion as to the relative merits of the two would be out of place at this time as investigation is not entirely concluded. At the present stage of development the type using arc voltage control seems to be somewhat favored as the actual arc efficiency has been found greater with this control. Operating difficulties may prove that the current control type, however, is more reliable. This is a question for those who operate these lights in service to decide. The two types have further very different methods of operating on "strike." This further introduces complications. The two essentials, reliability and high efficiency, make certain demands that neither type meets entirely as one essential is sacrificed to achieve a gain in the other and the result is a compromise between the two.

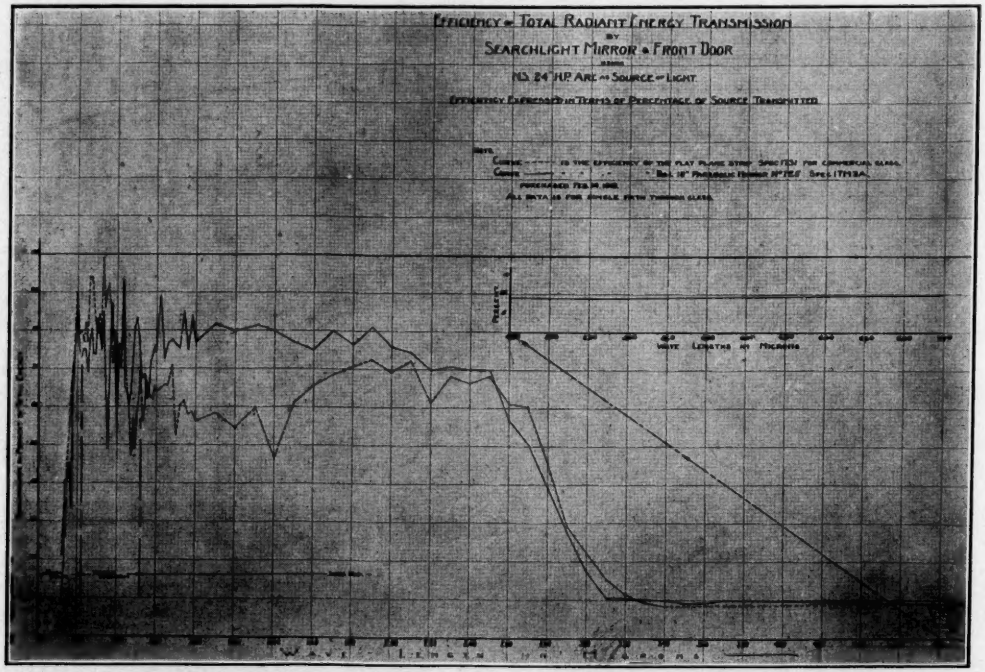
The necessity for the elimination of the copper-coated negatives, as before explained, resulted in a redesign of the negative head which increased arc efficiency. The advantages gained in efficiency were not all gained without some sacrifice, however, as permitting air burning of the negative materially reduced the life per trim of carbons. The original design gave a life per carbon trim of about four hours maximum. The necessary reduction to obtain the highest efficiency is slightly greater than 50 per cent of this but this reduction is not considered prohibitive for service conditions. The focal length of the mirror naturally limits an increase in length of the negative electrode itself. Here again a compromise in design is necessary and longer life with present electrodes cannot be obtained without loss in efficiency of illumination. Actual carbon consumption by weight in the high-powered arc is very little greater than that of the low power of the same nominal size and in some cases even less. Consumption by length is, however, much greater, due to the reduction in the diameter of the electrodes used with the high-powered arc. Carbon consumption bears no definite relation to intrinsic brilliancy of the crater.

Of course, in the preceding, nothing has been said of the searchlight mirrors. These have also undergone considerable development. The efficiency of the searchlight is after all dependent absolutely on the mathematical accuracy and physical perfection of its mirror. The mirror, however, is a complete subject in itself and hardly within the scope of this article. Nevertheless, it might be well to cover a few outstanding points.

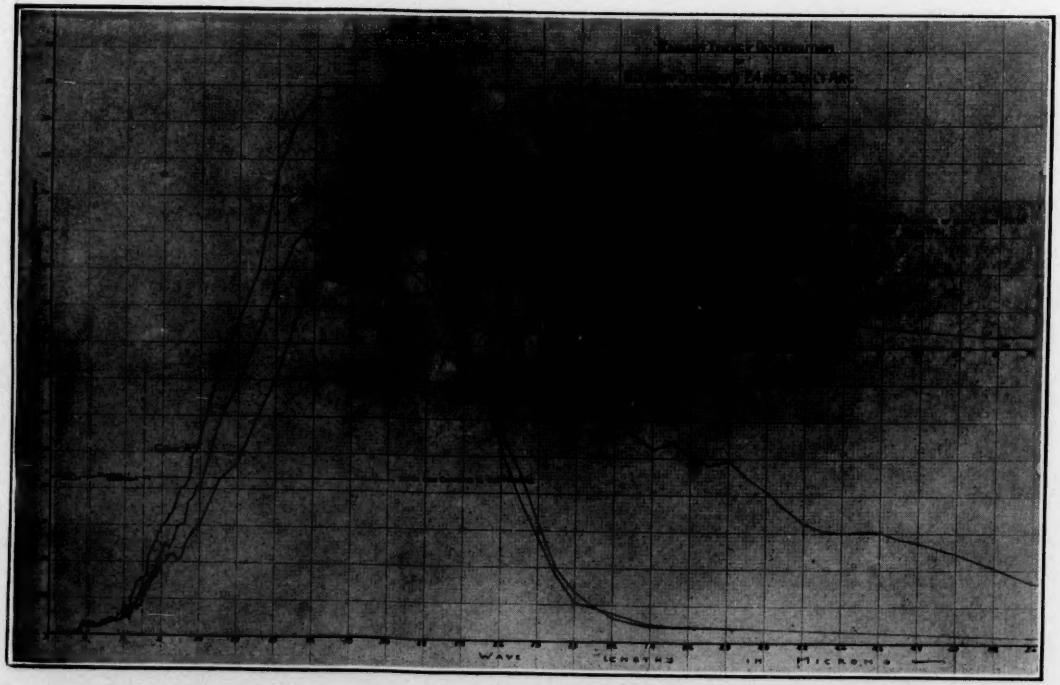
The function of the mirror is to collect the light from the source and project the rays in a beam as nearly parallel as possible. A development of the plano convex lens, known as the Fresnel lens, was one of the first methods used for this purpose and still is in use. In this case it was not a mirror, but took the place of the front door in the searchlight. It was



RADIANT ENERGY DISTRIBUTION IN THE VIOLET AND VISIBLE SPECTRUM SHOWING THE EFFECT ON THE ENERGY PRODUCED BY THE GLASS OF THE PROJECTION SYSTEM.


Efficiency of Total Radiant Energy Transmission
by
Searchlight Mirror & Front Door
using
No. 24 H.P. Arc as Source of Light.
Efficiency Expressed in Terms of Percentage of Source Transmitted
Note.
Curve ——— is the efficiency of the flat plane strip Spectrum for commercial glass.
Curve - - - - - - - - - - Bausch & Lomb Parabolic Reflector No. 7125 Spec. (7113 A.)
Performed Feb. 24, 1913.
All data is for single fifth Thickness glass.
Wave Lengths in Millimicrons

Curve Showing Relative Efficiency of Radiant Energy Transmission by the Glass of the Projection System Giving Efficiencies at Each Wave Length in Angstrum Units for the Entire Spectrum.



RADIANT ENERGY DISTRIBUTION CURVES SHOWING THE SELECTIVE EFFECT OF THE GLASS OF THE PROJECTION SYSTEM.

not found sufficiently adaptable and its ability to concentrate the light was not great enough, although it is still used in running lights, beacons, etc.

The Mangin mirror, a development by an officer of the French Army, then came into use. This invention first made the searchlight a really practical affair. This was a real mirror with a long focus and with the light source at this focus. The introduction of this mirror changed the system of projection from that by transmission as accomplished by the Fresnel lens to one of reflection. It is used today on smaller sizes of searchlights. It can be easily and cheaply made.

The Mangin mirror is of concave convex form and its two surfaces are a combination of parts of the surfaces of two spheres having radii of different lengths. By this means a reflector with an average focus between the centers of the two spheres was obtained. This gave a fairly parallel beam. The type is not known to have been produced in sizes larger than thirty-inch diameter. In this large size it weighed about ninety pounds and was about two inches thick at the edge and about half an inch thick at the center. Its weight thus imposed a limitation and its variable thickness was so excessive as to introduce great danger of breakage under high temperature conditions. The mechanical possibilities in its manufacture are its great advantage.

The parabolic lens finally solved the problem. It gave the nearly true parallel beam. At present the compensated parabolic mirror is used. The true parabolic mirror is used in astronomical telescopes and the concave surface is a true paraboloid of long focal length. The reflecting surface is a silver film on the inside or concave surface of a mathematically true glass paraboloid; the light coming directly to the polished reflecting surface and not passing through the glass.

In the case of the searchlight mirror the silver reflecting surface must, for protection from the arc, be placed on the

convex side of a paraboloid with both sides ground and polished. The light thus passes through the glass twice during reflection. The refraction resulting from the light passing twice through the glass, once in passing to the reflector and once in the reflection by the silver, would disturb the precision of the light projection, hence this effect must be compensated. The front and back surfaces of the glass cannot both be mathematically correct parabolas, and so the thickness of this type of mirror throughout its surface is also variable, being greater at the center than at the edges, though the amount of variation is very small. The mathematical formula for the forming of these surfaces is extremely complex and the apparatus used in the manufacture of the mirrors to produce the desired curvature is likewise complicated and extremely careful workmanship is absolutely essential in their production.

The quality of glass used is an important feature. It must contain no color, must be stable under heat, and must have a low co-efficient of expansion. It must have a very low energy absorption, five per cent of the available spectral energy, and about ninety-eight per cent efficiency of visible light transmission. The problem of satisfactory mirrors is a difficult one and to date only one American manufacturer has proved entirely successful in meeting it.

The use to which searchlights are put in the Naval service is particularly exacting. Mounted as they are, completely exposed to all weather conditions on super-structures, they are in addition subjected to vibration, shock and often to the rather corrosive effect of heated gases from the smokestacks. Particularly on destroyers they may even be exposed to the partial effects of shipping an occasional green sea. For an apparatus as delicate as a high powered lamp, this is very severe service.

Metals used in construction, therefore, must be exceptionally non-corrodible and the drums and electrical installation must be efficiently weather-proof. Junction boxes, scale lights, and instruments must be water tight and everything else where

it is practicable. The drum must be weather-proof against spray, fog and the elements in general, to protect the delicate parts of the lamp mechanism. It is difficult to make manufacturers understand the extreme severity of sea conditions on these lights. In service much can be done to maintain these lights at their highest efficiency by careful protection against the elements at all times.

The use of the high power arc in the Navy for certain purposes has many disadvantages. It will probably continue to be used, however, until a satisfactory substitute has been developed. For many other purposes it will always be desirable and many new applications in the military and naval fields have been found and will probably continue to be found as time goes on. This is especially so as new classes of offensive and defensive equipment are developed and adopted.

The Navy Department maintains a fully equipped laboratory at the Navy Yard, New York, for the complete testing of searchlights and all accessories. Much has been done there to stimulate and aid the development of the high power searchlight. Through making the many and complex tests for suitability of different types submitted by manufacturers, the Navy has acquired information in this line probably not equalled by any other organization.

FRICTION TESTS OF PROPELLER THRUST BEARINGS.

Kingsbury and Horseshoe Types Compared.

By H. A. Stevens Howarth, Associate Member, and Nelson Ogden.

The friction in a propeller thrust bearing can hardly be determined at all from an actual installation on shipboard. There are so many variables affecting the friction that very little confidence can be placed in data obtained in this way, even when the observations are taken with exacting care. However, it is important for ship operators to know the amount of friction developed in thrust bearings, not only because of the absolute loss of power, but also on account of the heating and consequent operating difficulties resulting from the loss of power. There is little data on the subject of friction in the multi-collar horseshoe type of thrust bearing, deductions from published information on journal bearing friction being misleading because of the difference in lubricating action. For the Kingsbury bearing, however, the friction can be predicted closely from theoretical considerations because of the positive and thorough lubricating action which results from its fundamentally correct design.

The experiments described in this paper were made to obtain reliable friction measurements on full-sized bearings of both the Kingsbury and the Horseshoe types, without regard to any theoretical considerations.

The testing machine used for the friction determinations is shown in Fig. 1. The shaft is supported by three steady bearings bolted to a bedplate and is driven by a large pulley. The Kingsbury bearing is mounted in a special housing which is

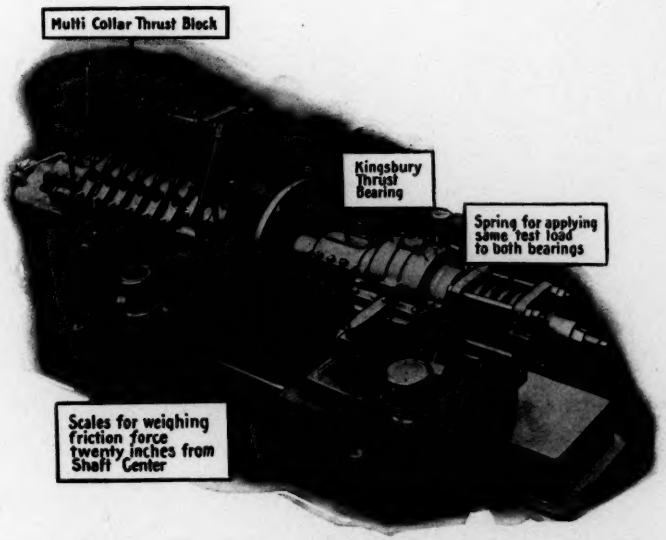
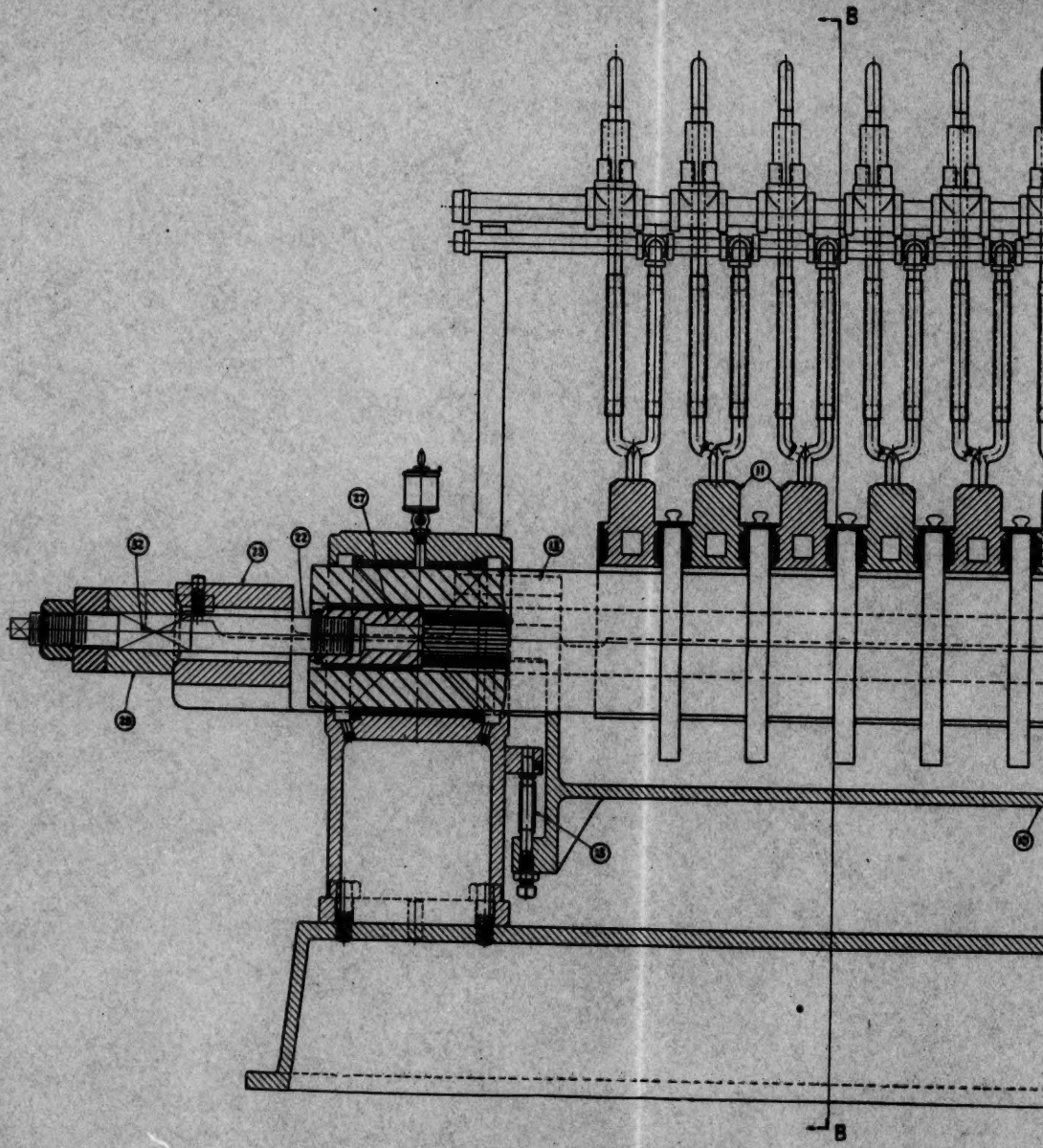

Multi Collar Thrust Block
Kingsbury Thrust Bearing
Spring for applying same test load to both bearings
Scales for weighing friction force twenty inches from Shaft Center

FIG. 1.—TESTING MACHINE FOR MEASURING FRICTION IN KINGSBURY AND HORSESHOE BEARINGS AT IDENTICAL LOADS AND SPEEDS.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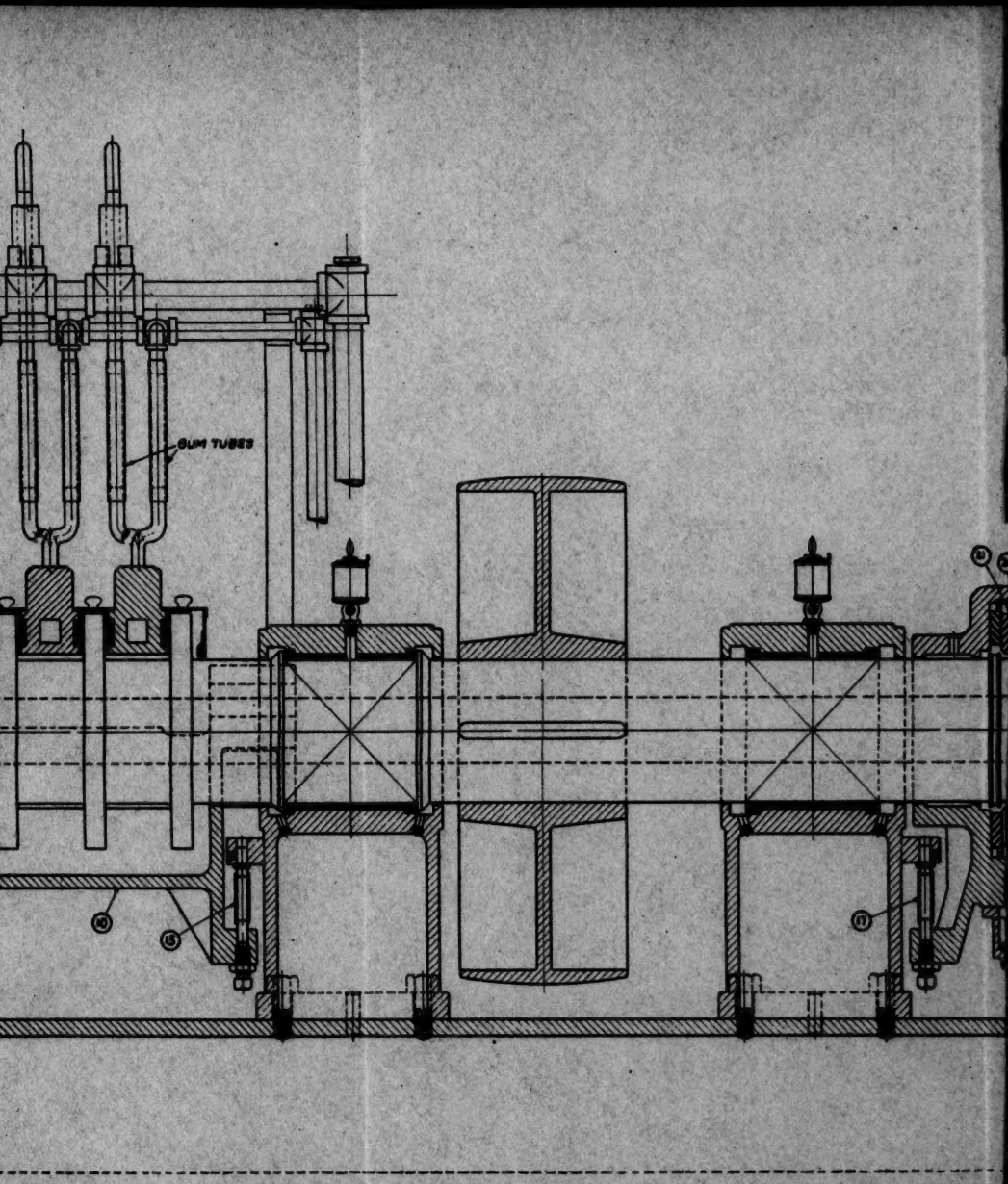


FIG. 2.—AXIAL SECTION OF TESTING MACHINE.

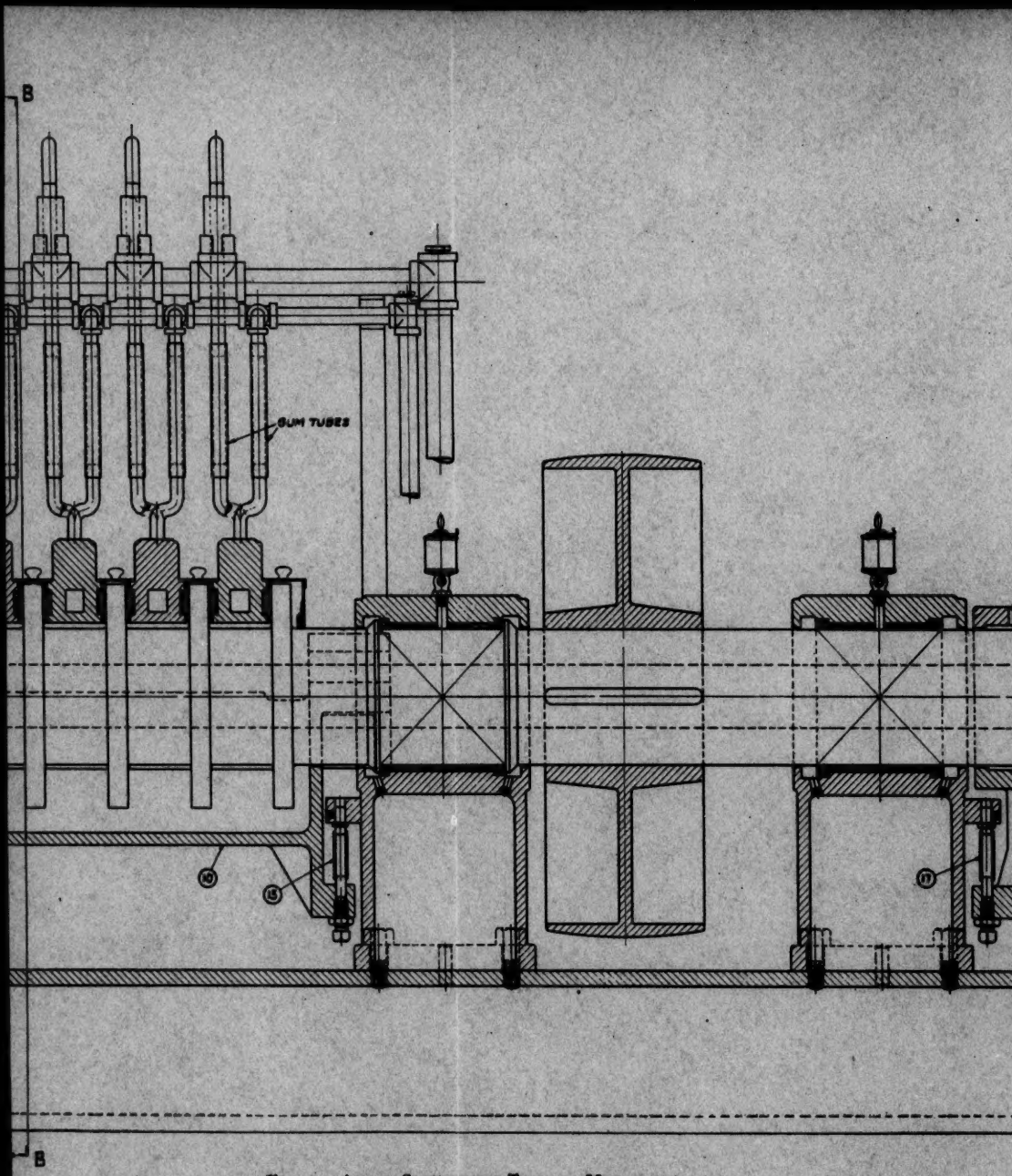
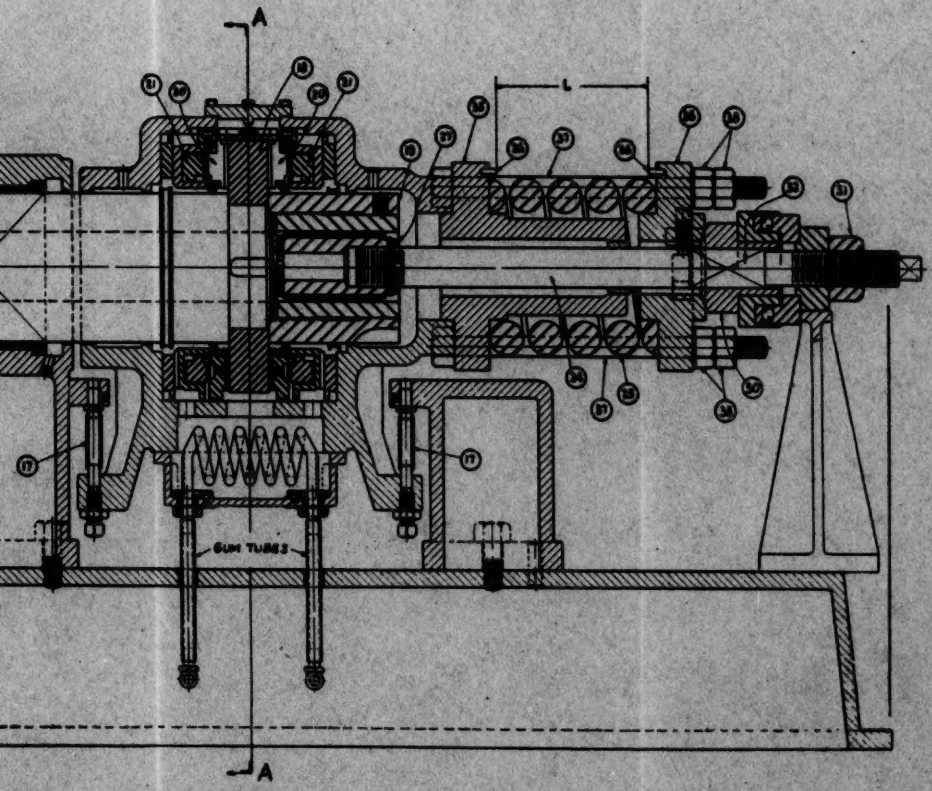


FIG. 2.—AXIAL SECTION OF TESTING MACHINE.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not bolted to the bedplate but is supported by spherical-ended pins (16) and is perfectly free to rotate through a small angle. This construction is shown in Figs. 2 and 3. The horseshoe bearing housing is similarly mounted on spherical-ended pins

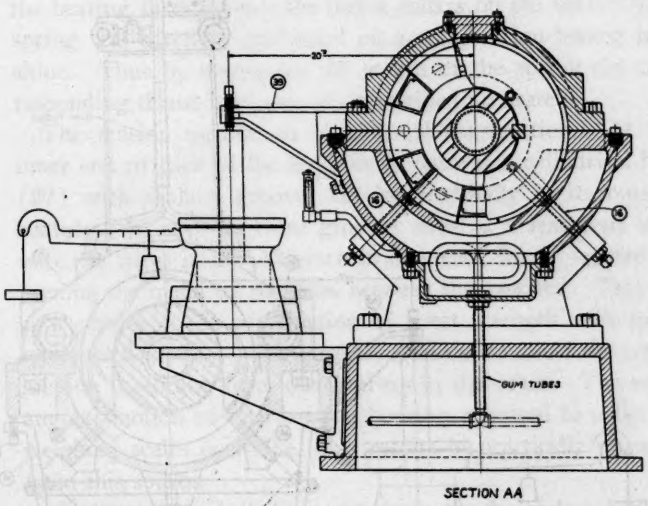


FIG. 3.—SECTION THROUGH KINGSBURY BEARING.

(14). Arms (39) and (40) project from each housing and the friction torque load is transmitted directly to the weighing scales shown in the illustration. Thus the thrust friction forces can be measured accurately for all loads and speeds. The friction of the steady bearings does not enter in any way.

In order to apply the thrust loads equally to the two bearings it was necessary to resort to the special construction shown in Fig. 2. The shaft is hollow and a flexible tension member passes through its center. On the horseshoe end this member is terminated in a stud (32) with a nut bearing against a spherical seated hub (29), which in turn bears against a yoke (23) bolted to the horseshoe bearing housing. At the end

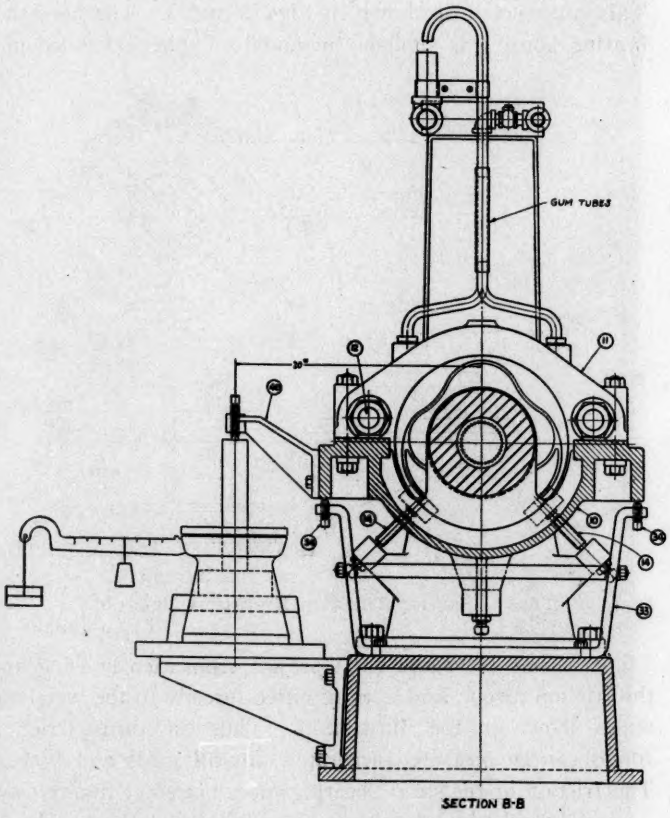


FIG. 4.—SECTION THROUGH HORSESHOE BEARING.

adjacent to the Kingsbury bearing the tension member terminates in a similar stud with a nut which transfers the load to the Kingsbury housing through a large spring. By setting up on the nut the spring is compressed and the same thrust load is applied to each of the two bearing housings which press the bearing faces against the thrust collars on the shaft. The spring was carefully calibrated on a compression testing machine. Thus by measuring the length of the spring the corresponding thrust load can be determined accurately.

The tension member is of unusual construction. At the inner end of each of the large studs there is a cylindrical hub (27) with shallow grooves cut longitudinally in its outside surface. In each of these grooves there is a tempered steel wire, 36 wires in all, .120-inch in diameter, firmly secured by peening the metal of the hubs between the grooves. This design results in the combination of great strength with great torsional flexibility. The latter is essential to avoid the transmission of torque from one bearing to the other. The small angular motion of each bearing housing, required to make the weighing scales operative, thus results in practically no error from this source.

Cooling water is led to and from the horseshoe and the Kingsbury bearings through thin pure gum tubes. These are placed in a vertical axial plane so that they do not disturb the friction torque readings in any way.

The design of the horseshoe thrust bearing was taken directly from drawings furnished by a well-known shipyard. The shaft diameter is 8 inches, the collars are 14 inches diameter, and the net bearing area of the six horse shoes is 365 square inches. This makes allowance for 30 square inches of the gross area lost in oil grooves. The mean diameter of the bearing surfaces is 11⅜ inches. The unit pressures usually applied to horseshoe bearings are in the neighborhood of 35 pounds per square inch, based on effective thrust loads. The rated capacity of the bearing on this basis is 12,800 pounds.

The Kingsbury bearing was designed for the same rated thrust capacity using the customary pressure of 250 pounds per square inch. The net bearing area of the segments is 51.4 square inches, the bearing diameter is 14 inches, and the mean diameter is 11½ inches. A double bearing, having segments on both sides of the thrust collar, was used in order to represent actual service conditions, although the thrust loads in these tests were applied in one direction only.

In order to guard against errors due to lack of balance in the bearing housings, the driving motor was made reversible and special counterweights were provided. These counterweights were heavier than the upward force developed when reversing. They were mounted at the extreme ends of the arms bearing on the weighing scales. By this means an average of forward and reverse readings was found.

The tests were run in July and August, 1922. The thrust loads ranged from 6,000 to 16,000 pounds, and the speeds were 62, 104, 166, and 262 R.P.M.

The oil used in the horseshoe bearing was a brand recommended by a shipbuilder for this bearing and is a kind very widely used for this class of service. Its viscosity at 100 degrees F. is about 650 seconds Saybolt. The bearing temperature was kept as low as possible by the cooling water, and the highest temperature reached, as determined by a thermometer installed in the oil bath, was 52 degrees C. For the Kingsbury bearing a lighter oil was selected having a viscosity of about 150 seconds Saybolt at 100 degrees F. It is a grade that is entirely suitable for the service. Cooling water was circulated through the Kingsbury bearing part of the time to keep the temperature at a proper point, and the highest temperature reached was 47 degrees C.

In selecting the oils, as well as in every other respect, the object kept in view was to reproduce marine service conditions as completely as possible. The machine itself was designed for very accurate readings and in its operation all adjustments,

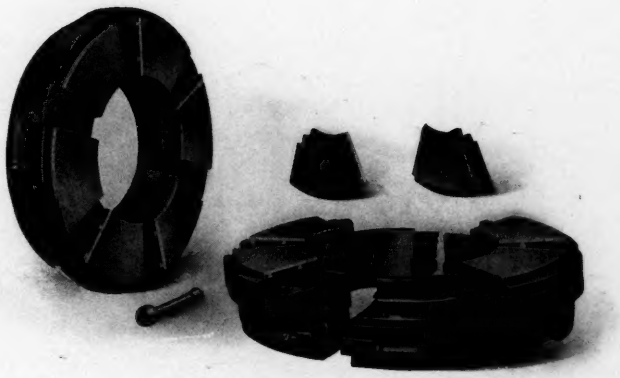


FIG. 5.—6-SHOE EQUALIZING KINGSBURY THRUST BEARING.

settings and readings were made with particular care. The horseshoes were squared up in place and the bearing faces were scraped to fit the collars. They were carefully adjusted from time to time during the tests, as the bearing warmed up, in order to obtain substantially equal load distribution. No such adjustment was necessary for the Kingsbury bearing, it being of the six-shoe, equalizing type shown by Fig. 5. At the end of the tests both thrust bearings were inspected and found to be in excellent condition. In this series of tests no attempt was made to test the bearings to destruction. On other occasions, however, it was found that the Kingsbury bearing remained in perfect condition when the babbitt dragged on the faces of the horseshoes.

In working up the test data it was evident at once that the oil temperatures have a very important influence upon the friction. Accordingly each observed torque load was reduced to its equivalent at a standard temperature arbitrarily assumed at 35 degrees C. (95 degrees F.). The correction curves which were used are shown by Fig. 6. They were developed from the tests themselves, averaging a number of long runs, during which the temperature change was considerable.

Figure 7 shows a sample of the plotting of the data. As indicated by the labels, the full sloping lines show weighing scale readings uncorrected for temperature, just as they were obtained from the testing machine. As the oil warmed up, the friction decreased, of course. The broken lines show the same readings corrected to 35 degrees C. Using these corrected lines, which were plotted for every test, a mean reading was selected and plotted on the Torque-Thrust charts, Figs. 8 and 9. From these curves were constructed the curves of relation between coefficient of friction, revolutions per minute, and bearing pressure, Figs. 10 and 11.

The friction coefficient curves present the results of the tests in the most generally serviceable form. It must be borne in mind, however, that each curve applies to a definite oil and

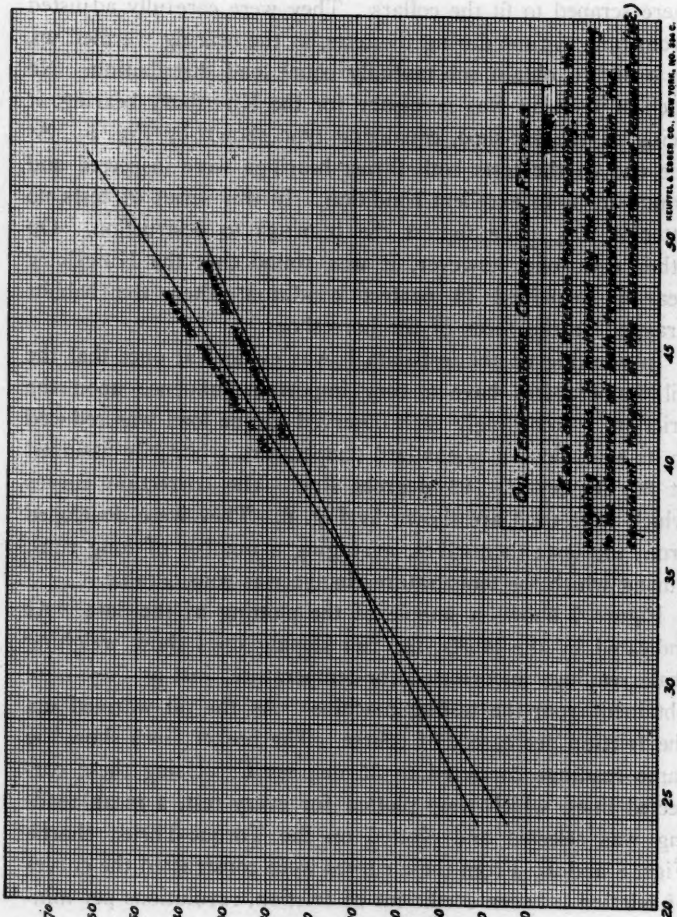


FIG. 6.—OIL TEMPERATURE CORRECTION FACTORS.

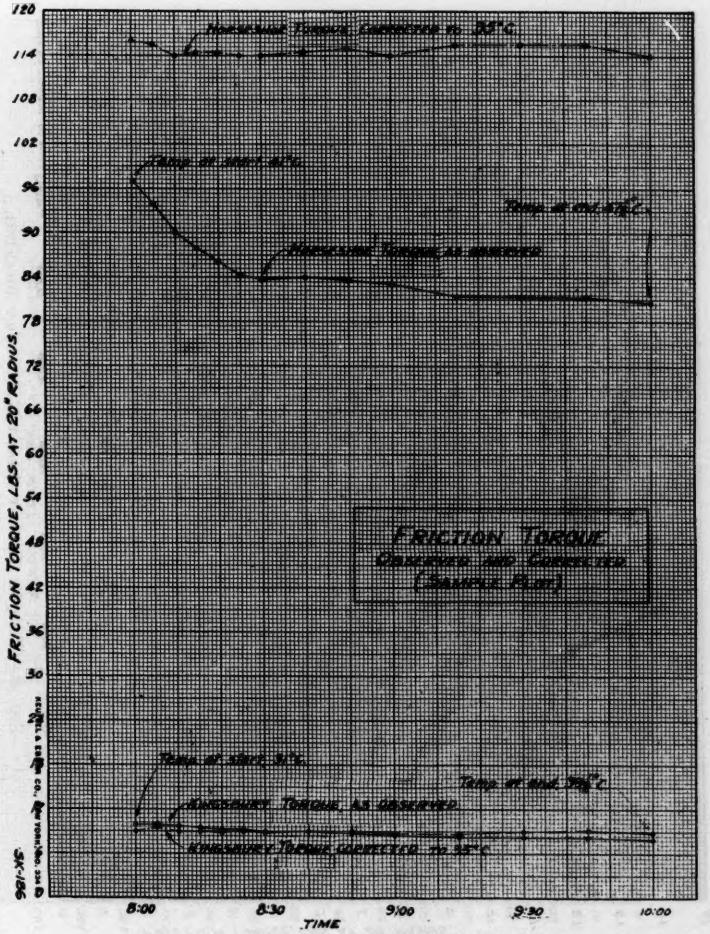


FIG. 7.—FRICTION TORQUE, OBSERVED AND CORRECTED. (SAMPLE PLOT.)

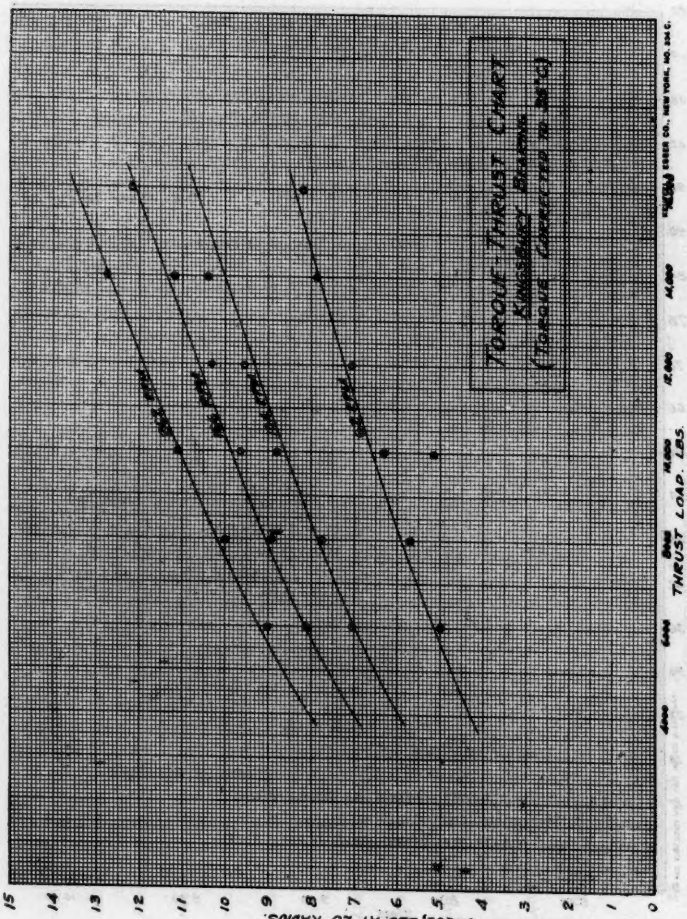


FIG. 8.—TORQUE THRUST CHART KINGSBURY BEARING.

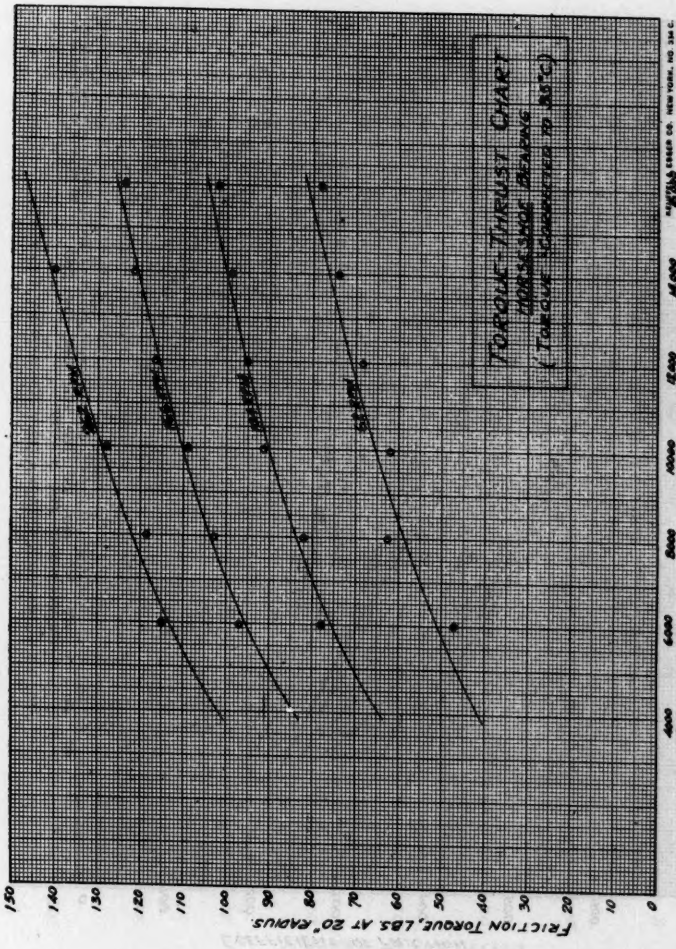


FIG. 9.—TORQUE THRUST CHART HORSESHOE BEARING.

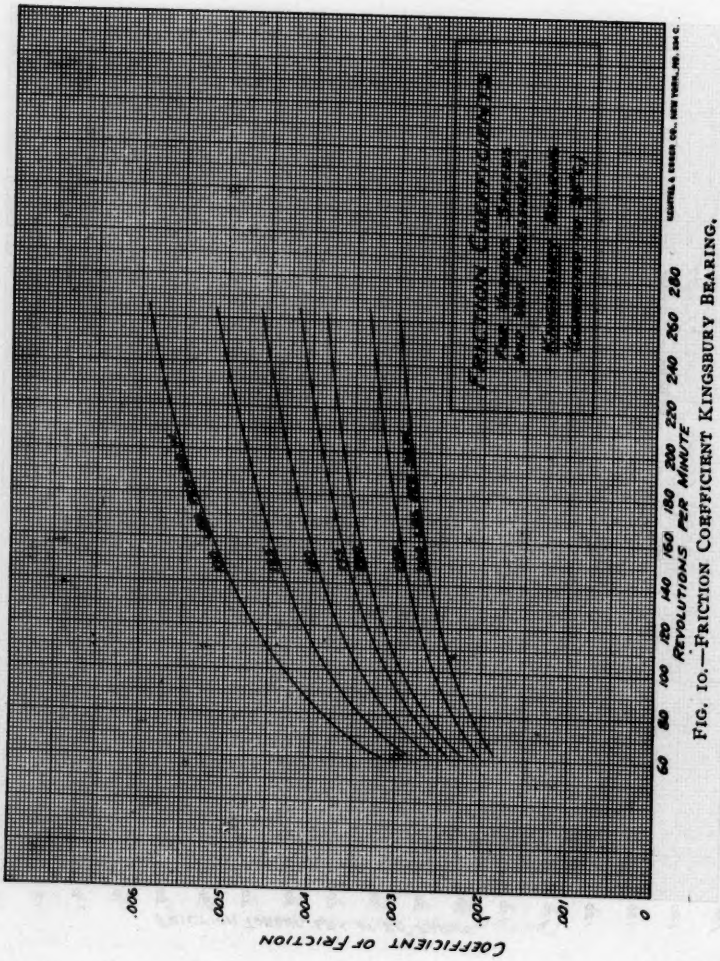


FIG. 10.—FRICTION COEFFICIENT KINGSBURY BEARING.

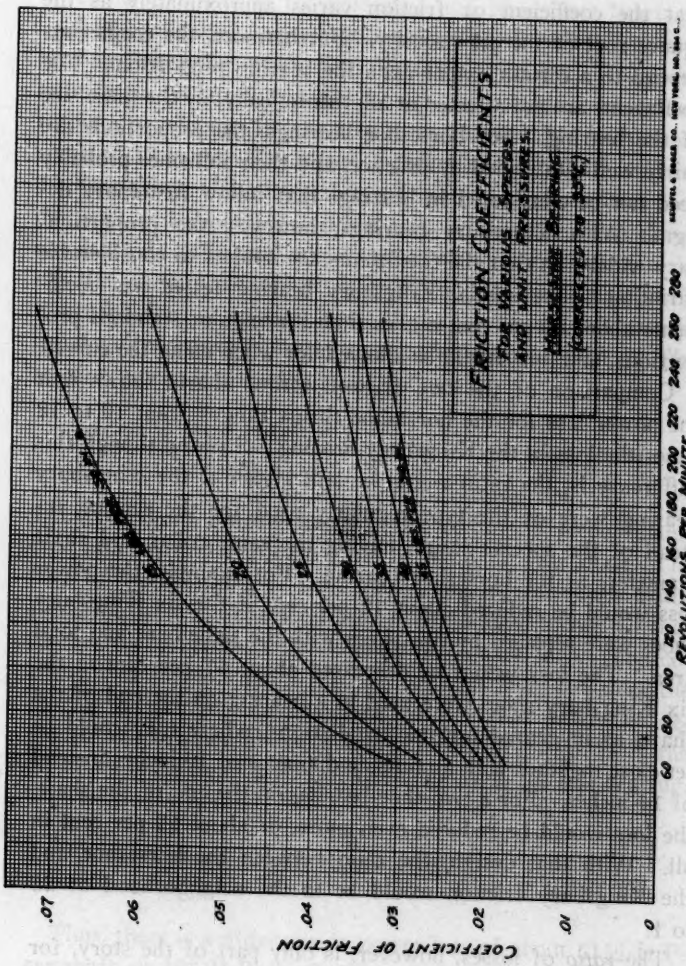


FIG. 11.—FRICTION COEFFICIENT HORSESHOE BEARING.

a definite running temperature. For other oils and other temperatures, suitable allowances can be made by remembering that the coefficient of friction varies approximately as the square root of the oil viscosity. Furthermore, the coefficients depend to a certain extent upon the bearing proportions. The coefficients are not affected at all, theoretically, by the mere size of the bearing. For horseshoe bearings, the proportions are fairly well established by practice, and their influence probably need not be considered in bearings that follow customary design. With Kingsbury bearings, however, the proportions vary widely, so that the coefficients are subject to considerable variation. The special Kingsbury bearing tested has coefficients about 20 per cent higher than bearings of standard proportions, and this should be allowed for when using the curves.

Comparison of the two friction torque charts shows that the power loss for the horseshoe thrust ranges from 10 to 12 times the loss in the Kingsbury bearing. Striking though this difference is, the ratio would be still higher if the Kingsbury bearing were of the type regularly used for the service, instead of the special design having the same collar and shaft diameters as the horseshoe bearing. For example, the power loss would be reduced about 12 per cent if the six Kingsbury shoes were replaced by two shoes with the same bore and total area. The loss would be decreased about 47 per cent if the six Kingsbury shoes were used in a bearing on the end of the shaft, as in geared turbine installations, where they could be set close together with an outside diameter of 10 inches instead of 14 inches. For a two-shoe bearing on the end of the shaft, the loss would be reduced even more, or about 50 per cent in all. With that design, the ratio between the horseshoe and the Kingsbury friction would be in the neighborhood of 22 to 1.

The *ratio* of losses, however, is only part of the story, for it is the absolute loss of power which is of controlling importance. The calculation in the following paragraph illustrates

the application of the test results to an actual problem, and is a numerical example of the possible saving represented by the difference in losses in the horseshoe and Kingsbury bearings.

There are two sister ships, one equipped with Kingsbury bearings and the other with horseshoe thrusts. They are twin-screw oil-burning passenger vessels, propelled by geared turbines developing 12,000 S.H.P. at 125 R.P.M., with a sea speed of 17 knots. The effective thrust load per shaft is about 70,000 pounds. The horseshoe bearing has 12 shoes, 26 inches diameter and about 16½ inches bore, so that the mean diameter is 21¼. The total area is about 2700 square inches, making the *average* bearing pressure 26 pounds per square inch. The coefficient of friction, from Fig. 11, is about .037, assuming the oil viscosity to be the same as in the test bearing. The friction power loss works out as follows:

$$\frac{70{,}000 \times .037 \times 21\frac{1}{4} \times \pi \times 125}{33{,}000 \times 12} = 54\frac{1}{2} \text{ horsepower.}$$

The Kingsbury bearing has a single collar, 25 inches diameter, attached to the forward end of the main gear shaft. The bore of the six shoes is 12½ inches, making the mean diameter 18¾ inches. The total area is 312 square inches, so that the bearing pressure is 224 pounds per square inch. The coefficient of friction, from Fig. 10, is about .003, but since the bearing proportions are standard, 20 per cent should be deducted as explained above, making the coefficient .0024 if the oil viscosity be again assumed the same as in the test bearing. The friction loss is:

$$\frac{70{,}000 \times .0024 \times 18\frac{3}{4} \times \pi \times 125}{33{,}000 \times 12} = 3.1 \text{ horsepower.}$$

Thus, there is a difference in power loss of about 51½ H.P. per shaft, or 103 H.P. in all. To calculate the money value of this power, during a year's service, assume the oil consump-

tion to be 1.1 pounds per S.H.P., the price of fuel to be $14 per ton, and the number of days at sea per year to be 250. Then the saving works out as follows:

$$\frac{103 \times 1.1 \times 24 \times 14 \times 250}{2240} = \$4{,}250, \text{ annual fuel saving.}$$

The various assumptions made are subject to revision, of course, to suit the various conditions of actual service, but it is believed the calculation will serve as a basis for determining the losses for any given installation.

TEST OF MAIN CONDENSERS OF BATTLESHIPS *TENNESSEE* AND *MARYLAND*.

By Joseph B. Lincoln and O. A. Van Denburgh, Jr.,
Mechanical Engineers.

Due to the fact that the condensers of the U. S. S. *Tennessee* were of Lovekin design, a type new to the Naval Service, it was desired that a complete test be made upon them for the purpose of obtaining data from which their performance could be compared with that of the usual Navy type condenser. This required that another similar test be conducted on a typical Navy type unit and for this purpose the U. S. S. *Maryland* was selected.

These two ships were practically duplicates. Both were electric drive. The *Maryland's* power plant consisted of two General Electric Company turbines rated at 11,000 K. W. each at 2030 R.P.M. The *Tennessee* had two Westinghouse turbines rated at 10,000 K. W. each at approximately the same speed. In both cases the turbines, each with its individual condenser, were located in separate engine rooms, forward and aft. All tests were conducted in the after engine rooms.

As these tests involved measurement of the quantity of steam condensed and as it was desired to obtain data at various ship speeds, both tests were made during the official trial of each respective ship, thus avoiding duplication of work and considerable expense.

DESCRIPTION OF MAIN CONDENSERS OF U. S. S. "TENNESSEE."

The main condensers of the U. S. S. *Tennessee* were of the Lovekin type, having a total cooling surface of 11,616 square feet. There were 6604 $\frac{5}{8}$-inch tubes, each 10 feet 10$\frac{1}{2}$ inches long. The path of the circulating water was divided into two

passes, an upper and a lower. The upper pass had 3323 tubes, and the lower pass had 3281 tubes. The circulating water entered at the lower pass, and was discharged from the upper pass. The inside diameter of the shell was 9 feet and the length between tube sheets was 10 feet 9 inches. There were separate air ejection and condensate suctions. The condenser was situated directly below the main generator turbine, and had two large exhaust nozzles connected to the turbine exhaust casing by an expansion flange. There was a motor driven centrifugal pump, used as a main circulating pump designed to pump 19,000 gallons per minute against a head of 30 feet. The diameter of the inlet and discharge piping to the condenser was 28 inches.

The particular features claimed for the Lovekin type condenser may be stated as follows:—

(*a*) A maximum inlet area to the tubes and between the tubes.

(*b*) Pitching the tubes so that the steam lanes are in the direction of flow of steam.

(*c*) Reducing the depth of the path of the steam flow.

(*d*) Arranging for the condensate to be drawn from the upper bank of tubes at its maximum temperature, and also keeping this condensate from passing over and wetting the lower tubes.

(*e*) Providing a liberal cooling chamber for the air and non-condensible vapors.

Details of the Lovekin type condenser are shown on Figure I. The arrangement of tubes and baffles in the main condensers of the U. S. S. *Tennessee* varied slightly from that shown. The features noted in the preceding paragraph can readily be observed.

The air cooling arrangement is of interest. The air is drawn under the middle series of baffles and into the narrow vertical compartment through numerous one-inch diameter holes. The path of the air is then downward into the en-

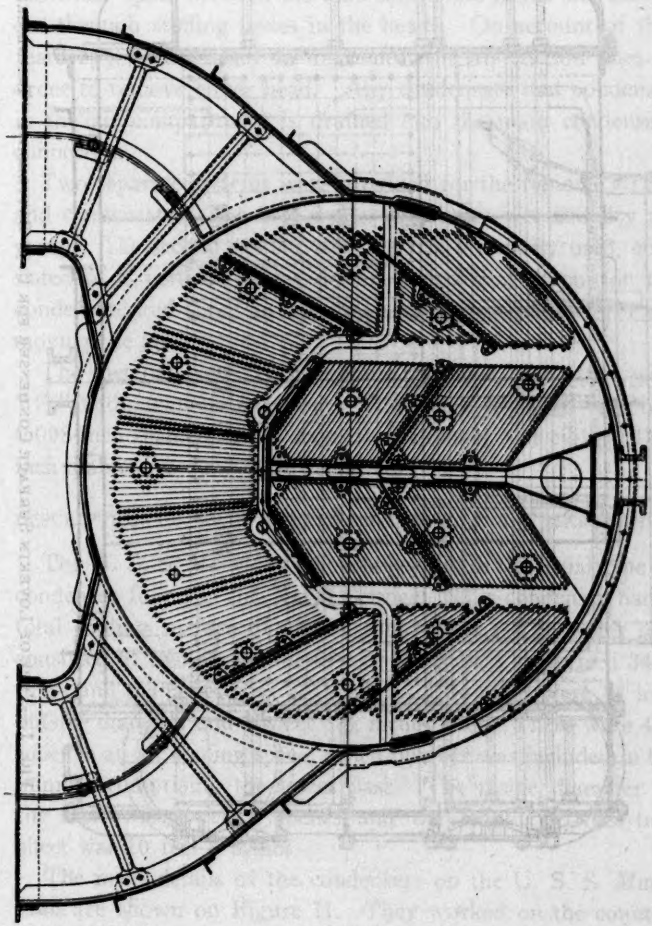


FIG. 1.—TYPE OF LOVEKIN SURFACE CONDENSER FOR U. S. BATTLESHIP "TENNESSEE."

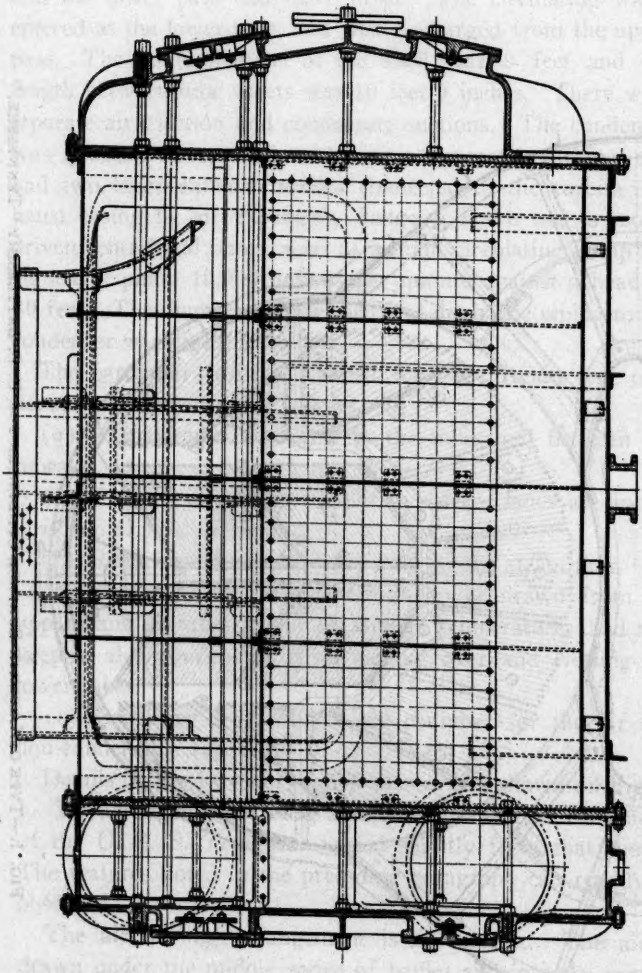


FIG. I.—TYPE OF LOVEKIN SURFACE CONDENSER FOR U. S. BATTLESHIP "TENNESSEE."

larged section of the compartment at the bottom. The compartment extends the full distance between tube sheets. The air is withdrawn from this at both ends of the condenser. The fittings provide for the removal of the air passed through the water space between the tube sheets and heads and thence out through stuffing boxes in the heads. On account of this feature, it is necessary to disconnect the air suction lines in order to remove either head. Any condensate that condenses in the air compartment is drained into the main condensate connection.

Two separate systems were provided for the removal of air and condensate. The first was a twin beam wet and dry air pump. The second system, and the one ordinarily used, consisted of a vertical, motor driven, centrifugal pump for the condensate and three air ejectors arranged in parallel for removing the air.

The ejectors were Westinghouse LeBlanc, two stage, size "G," without intercondensers. They were equipped with one 0.098-inch diameter first stage steam nozzle and eight 0.138-inch diameter second stage steam nozzles.

DESCRIPTION OF MAIN CONDENSERS OF U. S. S. "MARYLAND."

The U. S. S. *Maryland* was equipped with the usual type of condenser found in the Naval Service. This condenser had a total cooling surface of 12,050 square feet. The water side consisted of two passes, the first or lower pass comprised 3499 tubes and the upper pass 3515 tubes. The tubes were $\frac{5}{8}$ inch outside diameter and 10 feet $8\frac{7}{8}$ inches long. There were 438 tubes in an air cooling space, which number was included in the number comprising the lower pass. The inside diameter of the shell was 9 feet 3 inches and the length between tube sheet was 10 feet 6 inches.

The main details of the condensers on the U. S. S. *Maryland* are shown on Figure II. They worked on the counter-current principle and on the so-called dry system. A single

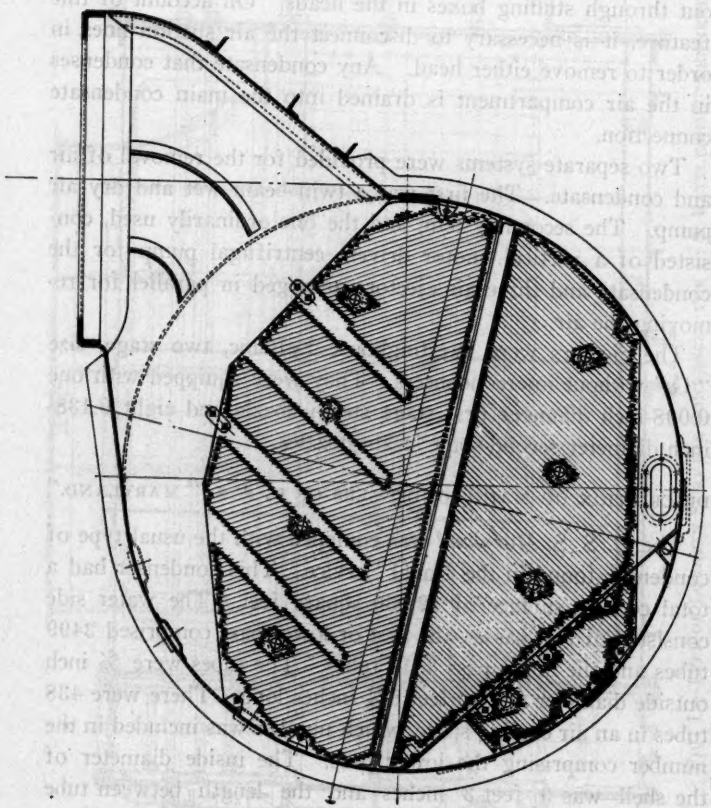


FIG. II.—NAVY TYPE SURFACE CONDENSER FOR U. S. BATTLESHIP "MARYLAND."

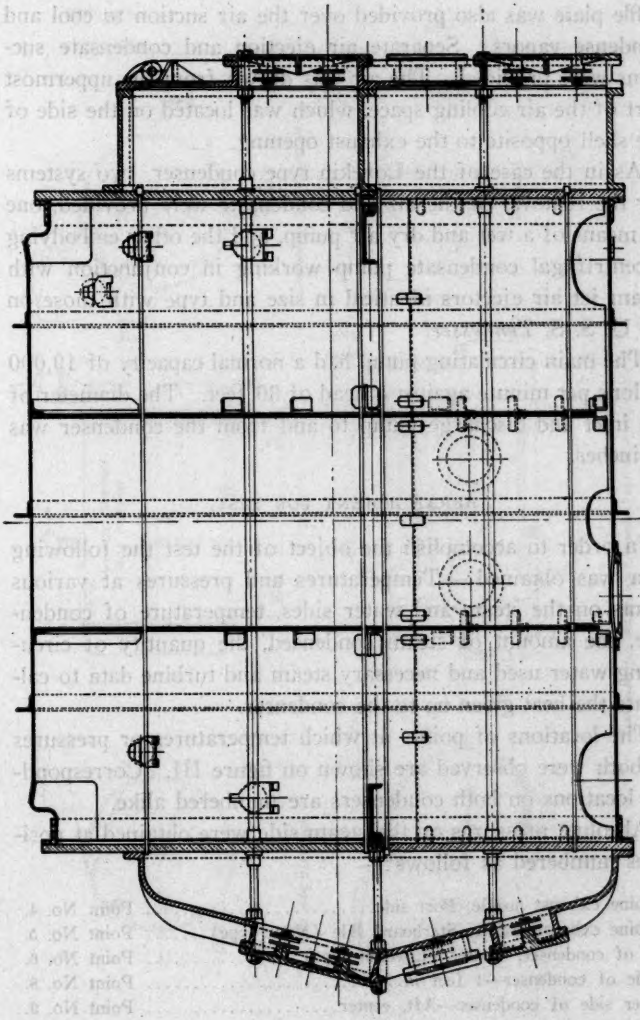


FIG. II.—NAVY TYPE SURFACE CONDENSER FOR U. S. BATTLESHIP "MARYLAND."

exhaust opening was provided on the top of the shell. A baffle plate was also provided over the air suction to cool and condense vapors. Separate air ejection and condensate suctions were provided. The air was drawn from the uppermost part of the air cooling space, which was located on the side of the shell opposite to the exhaust opening.

As in the case of the Lovekin type condenser, two systems for the removal of the air and condensate were provided, one by means of a wet and dry air pump, and the other embodying a centrifugal condensate pump working in conjunction with steam jet air ejectors identical in size and type with those on the U. S. S. *Tennessee*.

The main circulating pump had a normal capacity of 19,000 gallons per minute against a head of 30 feet. The diameter of the inlet and discharge pump to and from the condenser was 28 inches.

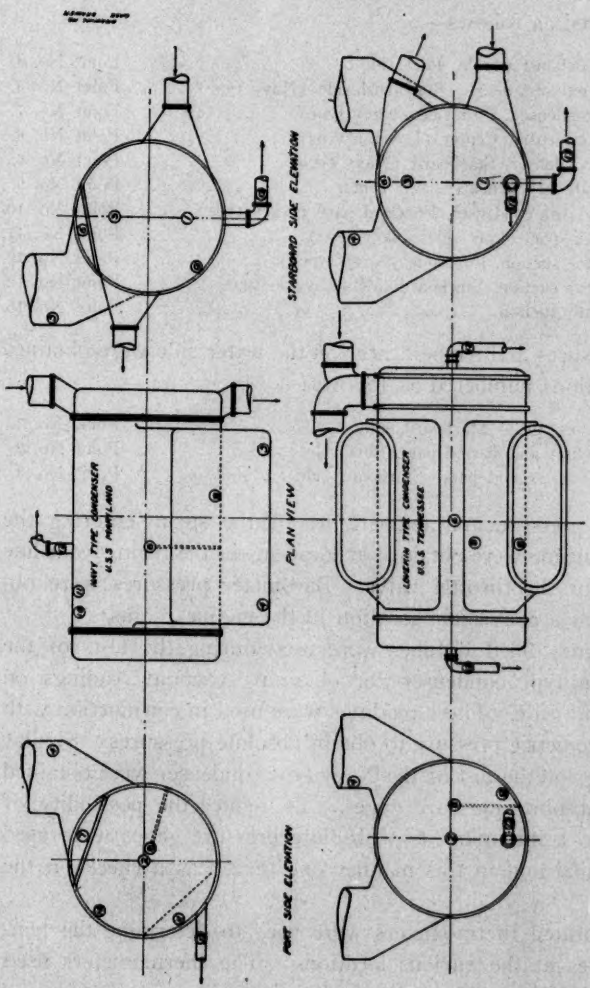
ARRANGEMENT FOR TEST.

In order to accomplish the object of the test the following data was obtained:—Temperatures and pressures at various points on the steam and water sides, temperature of condensate, the amount of steam condensed, the quantity of circulating water used and necessary steam and turbine data to calculate the heat given up to the condenser.

The locations of points at which temperatures or pressures or both were observed are shown on figure III. Corresponding locations on both condensers are numbered alike.

Absolute pressures on the steam side were obtained at positions numbered as follows:—

Turbine exhaust nozzle, Port side	Point No. 4.
Turbine exhaust nozzle, Starboard side (Navy type)	Point No. 5.
Top of condenser, Center (Lovekin type)	Point No. 6.
Inside of condenser—4 feet in	Point No. 8.
Lower side of condenser—Aft, center	Point No. 9.
Air cooler space—Port side (Navy type)	Point No. 11.
Air ejector Suction	Point No. 12.



DIAGRAMMATIC ARRANGEMENT SHOWING LOCATION OF TEST POINTS ON NAVY AND LOKERN TYPE CONDENSERS

FIG. III.

Temperatures on the steam side were obtained at positions numbered as follows—

Turbine exhaust nozzle, Port side	Point No. 4.
Turbine exhaust nozzle, Starboard side (Navy type)	Point No. 5.
Top of condenser, Port side (Navy type)	Point No. 7.
Top of condenser, Center (Lovekin type)	Point No. 6.
Top of condenser, Starboard (Navy type)	Point No. 6.
Lower side of condenser, Aft center	Point No. 9.
Upper side of condenser, Forward part (Navy type)	Point No. 10.
Air cooler space, Port side (Navy type)	Point No. 11.
Air ejector suction, Port side (Navy type)	Point No. 12.
Air ejector suction, Starboard side (Lovekin type)	Point No. 12.
Condensate suction	Point No. 13.

Pressures and temperatures on the water side were obtained at positions numbered as follows:—

Inlet to first pass—Starboard side	Point No. 1.
Between first and second pass, Port side	Point No. 2.
Outlet, from second pass—Starboard side	Point No. 3.

The pressure and temperature of the steam entering the main turbine were obtained at locations in the main steam line ahead of the throttle valve. Barometer pressures were obtained at a convenient location in the engine rooms.

Mercury filled U-tubes were used during the tests of the Lovekin type condenser for obtaining vacuum readings on the steam side. These readings were used in conjunction with the barometric pressure to obtain absolute pressures. Similar readings on the test of the Navy type condenser were obtained from absolute pressure gages. To reduce the possibility of error to a minimum, two absolute pressure gages were used in parallel and in this manner one served as a check on the other.

Calibrated thermometers were used to determine the temperatures at the various locations. The thermometers used for obtaining temperatures on the steam side of the condenser were inserted through exposed bulb connections and the bulb was wrapped with a piece of thin aluminum sheet to eliminate

TYPE				
RUN NUMBER				
DATE, 1931				
BAROMETRIC PRESSURE, IN OF HG.				
HORSEPOWER DEVELOPED BY TURBI				
SQUARE FEET OF COOLING SUR				
STEAM CONDENSED PER HOUR, P				
HEAT TO CONDENSER, B.T.U PER				
QUALITY OF STEAM ENTERING C PER CENT DRY				
STEAM DATA	ABSOLUTE PRESSURES IN CONDENSER, IN. OF HG.	TURBINE EXHAUST NOZZLE	PORT	
			PORT	
			STARBOA	
			POINT	
		TOP OF CONDENSER	CENTE	
			POINT	
		MIDDLE OF CONDENSER	FOUR FE	
			POINT	
		LOWER SIDE AFT	CENTE	
			POINT	
		AIR COOLER BANCE	PORT	
			POINT	
		EJECTOR SUCTION	PORT	
			POINT	
	STEAM TEMPERATURES IN CONDENSER, DEG. FAHR.	TURBINE EXHAUST NOZZLE	PORT	
			POINT	
			STARBOA	
			POINT	
		TOP OF CONDENSER	PORT	
			POINT	
			CENTE	
			POINT	
			STARBOA	
			POINT	
		LOWER SIDE AFT	CENT	
			POINT	
		UPPER SIDE FORWARD	PORT	
			POINT	
		AIR COOLER BANCE	PORT	
			POINT	
		AIR EJECTOR SUCTION	PORT	
			POINT	
			STARBOA	
			POINT	
		CONDENSATE	STARBOA	
			POINT	
WATER DATA	CIRCULATING WATER, GAL. PER MI			
	WATER TEMPERATURES IN CONDENSER, DEG. FAHR.	INLET	STARBOA	
			POINT	
		BETWEEN PASSES	PORT	
			POINT	
		OUTLET	STARBOA	
			POINT	
		TEMPERATURE RISES		FIRST
				SECOND
				TOTA
	WATER PRESSURE DROPS, LBS. PER SQ. IN. GAUG	FIRST PASS		
		SECOND PASS		
		TOTAL		
MEAN TEMPERATURE DIFFERENCE STEAM AND WATER, DEG. FAHR.				
HEAT TRANSFER COEFFICIENT, B.T.U.				
COOLING SURFACE PER HOUR PER SQ. FOOT				

TENNESSEE A

OBSERV

TABLE I

		LOVEKIN				NAVY				
		1	2	3	4	1	2	3	4	5
		18 MAY	19 MAY	19 MAY	20 MAY	11 NOV	13 NOV	15 NOV.	16 NOV	16 NOV
ESSURE, IN OF HG.		30.11	30.35	30.40	30.45	30.33	30.38	30.03	30.04	29.97
LOPED BY TURBINE		3066	10,731	11,298	15,378	10,312	10,589	10,943	16,403	15,581
F COOLING SURFACE		11,616				12,050				
D PER HOUR, POUNDS		41,513	108,672	118,284	174,740	101,344	105,864	108,320	166,419	157,819
ER, B.T.U PER POUND		991.3	912.2	915.3	931.6	928.7	933.4	932.2	934.0	927.8
PT ENTERING CONDENSER		99.21	86.51	87.45	88.69	86.65	86.66	86.41	86.35	86.72
HAUST NOZZLE OF CONDENSER	PORT SIDE POINT NO.4	0.29	0.65	1.07	1.17		0.92	1.09	1.30	1.34
	STARBOARD SIDE POINT NO.5						0.91	1.06	1.26	1.25
	CENTER POINT NO.6	0.40	0.65	1.08	1.14					
	FOUR FEET IN POINT NO.7	0.76	0.83	1.28	1.04		0.95	1.06	1.14	1.18
	CENTER POINT NO.8	0.63	0.75	1.23	1.02		0.89	1.13	1.05	1.09
	PORT SIDE POINT NO.11						0.95	1.12	1.20	1.23
	PORT SIDE POINT NO.12	0.55	0.71	1.19	1.08		0.93	1.08	1.02	1.04
	PORT SIDE POINT NO.1	68.9	77.3	88.8	87.5	88.2	84.2	87.3	96.2	94.9
	STARBOARD SIDE POINT NO.5					88.3	84.0	86.8	94.2	94.1
	PORT SIDE POINT NO.7					84.0	77.4	81.2	86.0	88.1
	CENTER POINT NO.6	87.7	80.0	89.2	87.5					
	STARBOARD SIDE POINT NO.6					84.0	76.7	80.7	85.6	86.1
	CENTER POINT NO.8	68.2	74.6	87.0	83.9	83.2	75.6	80.2	81.4	83.6
	PORT SIDE POINT NO.10					84.0	77.1	81.8	85.5	86.1
	PORT SIDE POINT NO.11					57.8	55.5	57.1	63.9	62.5
	PORT SIDE POINT NO.12					71.7	64.1	68.3	67.5	71.2
	STARBOARD SIDE POINT NO.13	51.1	61.9	77.4	70.2					
	STARBOARD SIDE POINT NO.13	61.8	74.8	86.5	83.1	62.7	55.3	55.5	58.8	67.4
ER, GAL. PER MIN.		11,000	14,600	14,000	15,700	18,000	17,600	18,500	19,200	18,400
LET	STARBOARD SIDE POINT NO.1	47.6	48.5	60.4	50.6	50.6	47.4	47.9	47.1	52.1
HEIN BER	PORT SIDE POINT NO.2	47.7	49.8	64.2	55.0	50.6	47.4	47.9	47.7	54.1
TLET	STARBOARD SIDE POINT NO.3	54.8	63.3	76.9	70.3	62.3	58.8	53.8	64.0	68.6
	FIRST PASS	0.1	1.3	3.8	4.4	0.0	0.0	0.0	0.6	2.0
	SECOND PASS	7.1	13.5	12.7	15.3	11.7	11.4	11.9	16.3	14.5
	TOTAL	7.2	14.8	16.5	19.7	11.7	11.4	11.9	16.9	16.5
FIRST PASS		1.52	3.14	2.90	3.78	4.82	3.97	3.82	4.82	4.68
SECOND PASS		1.33	2.55	2.45	3.14	4.64	5.12	5.27	4.79	4.82
TOTAL		2.85	5.69	5.35	6.92	9.46	9.09	9.09	9.61	9.50
URE DIFFERENCE BETWEEN R, DEG. FAHR.		17.15	20.52	18.44	21.55	26.93	23.13	26.71	28.21	24.74
DEPARTURE, B.T.U. PER SQ.FT. R HOUR PER DEG. FRAH TEMP. DIFF.		206.57	415.89	505.87	570.94	290.61	351.18	313.79	456.51	491.16

ESSEE AND MARYLAND CONDENSER TESTS
OBSERVED AND DEDUCED DATA

as much as possible the effect of radiation to the colder surrounding surfaces. The thermometers used for obtaining water temperatures were also inserted through exposed bulb connections.

Water pressures were obtained by means of mercury columns. Actual pressures and pressure drops were obtained after correction for elevation above an arbitrary datum level.

METHOD OF TEST.

The tests were made during the endurance trials of the ships at sea. After the start of each trial a sufficient length of time was allowed to elapse in order that steady conditions might be obtained. Readings were then taken by two observers at intervals of about one-half hour during the remainder of the run or until an average set of readings was obtained.

RESULTS OF TESTS.

The average data and deduced results of tests are shown in Table 1. The power developed by the turbines was obtained from official data with the aid of motor and generator efficiencies furnished by the manufacturers of this apparatus.

The heat in the steam entering the condensers was obtained by deducting the heat equivalent of work performed by the turbine with due consideration of frictional and radiation losses, from the heat of the steam supplied to the turbine. This latter quantity was obtained from the temperature and pressure observations on the steam entering the turbine. The amount of steam consumed was obtained from official data taken during the trials. The heat content of the exhaust steam together with the pressure at exhaust were used to calculate the quality of this steam as given in the table.

The quantity of circulating water used was obtained from performance curves of the circulating pumps with the aid of pump data observed during the runs.

DISCUSSION OF RESULTS.

Although it was impossible to make as many runs as was originally contemplated, it is believed that a sufficient number were made to obtain a fair comparison of the performance of the Lovekin and Navy type of condensers.

Particular attention is invited to the coefficients of heat transfer obtained with the condensers. With equivalent quantities of steam condensed, the Lovekin type condenser showed an increase of from 15 to 35 per cent over that obtained with the Navy type condenser. This increase in heat transfer was attained with a lower quantity of cooling water, and since the cross sectional area of the tubes is approximately the same in both cases, the velocity of this water was lower.

With corresponding steam quantities the average vacuum obtained was higher in the Lovekin than in the Navy type of condenser. Of course, air leakage would materially affect these valves, but it is very probable that only minor leaks, if any, were present, as both condensers were practically new.

The temperature of the condensate from the Lovekin type condenser was invariably nearer the average temperature of the steam space than in the Navy type condenser. In no case is the condensate from the Lovekin type condenser cooled below the temperature of the overboard delivery; whereas with the Navy type condenser, the temperature of condensate is approximately equal to or below the overboard delivery temperature.

The mean temperature difference between the steam and water was in every instance higher in the Navy than in the Lovekin type of condenser. Even at the lower temperature difference and a lower water velocity, the heat transfer was greater in the Lovekin condenser.

In both condensers most of the heat was absorbed in the second pass of the cooling water. Even at the maximum rate of steam flow, there is not more than 22 per cent of the heat absorbed in the first pass of either condenser.

The pressure drops on the water side of the condensers are practically equal when equal quantities of water are considered.

The air cooling space is apparently equally effective in both condensers when the effect of all influencing conditions is considered.

CONCLUSIONS.

The main points to be brought out as a result of the tests of the Lovekin and Navy type condensers may be stated as follows:—

The cooling surface in the Lovekin type condenser is from 15 to 35 per cent more efficient than that in the Navy type condenser.

The temperature of the condensate from the Lovekin type condenser was within a few degrees of the temperature corresponding to the vacuum in the condenser, while in the Navy type condenser the condensate temperature was from 20 to 30 degrees lower.

The air cooling arrangements in both condensers are apparently equally effective.

In general it would seem that the Lovekin type of condenser is superior to the usual type of condenser used in the Naval Service.

PROTECTING GASOLINE STORAGE TANKS FROM EVAPORATION.

By R. Z. Kirkpatrick, Chief Hydrographer, Panama Canal.

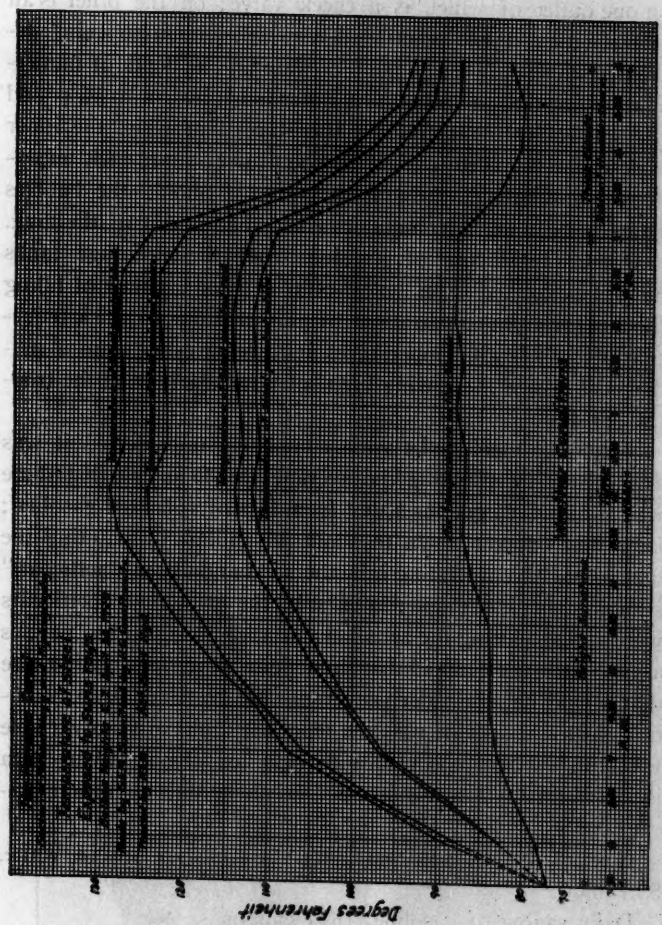
The Panama Canal has two 220,000-gallon gasoline steel storage tanks. Owing to the distance from the oil fields, these storage tanks are necessary, mostly as a reserve, but gasoline is occasionally drawn from them. As first built, exposed to the sun, the tanks gave a considerable loss from evaporation.

The weather conditions in the neighborhood of the tanks, are:

Maximum air temperature (shade)	95 degrees F.
Minimum air temperature	68 degrees F.
Mean Maximum temperature	86 degrees F.
Mean Minimum temperature	76 degrees F.
Average annual temperature	80 degrees F.

It is seen that these are relatively small temperature changes.

The tanks are 40 feet in diameter and 24 feet high. They are connected by pipe lines to and from the oil cribs for taking gasoline from tankers or delivery to ships. The tanks and lines are especially caulked for volatile oils, so that there is a minimum escape of gas at rivets, valves, etc. However, to prevent straining of tank plates when filling from a tanker, or collapse from outside air pressure, when drawing considerable quantities of gasoline, an air intake and a vent were necessary. Air is admitted through a riser, extending a few inches above the top of the roof, and provided with a check valve opening inwards, which allows air to pass into the interior of the tank


Degrees Fahrenheit

in case of falling of inside pressure. To provide for raising the inside pressure a riser was provided with a tee at the top, on one outlet of which is a check valve; on the other is an elbow with a nipple pointing downward, with its opening depressed sufficiently into a bowl of water. The depth of immersion is such that the column of water is equivalent to ¼ pound per square inch. It follows, therefore, that when the inner tank vapor pressures exceed ¼ pound per square inch uncondensed gas bubbles through the water. Should the bubbles ignite fire cannot be communicated to the interior of the tank.

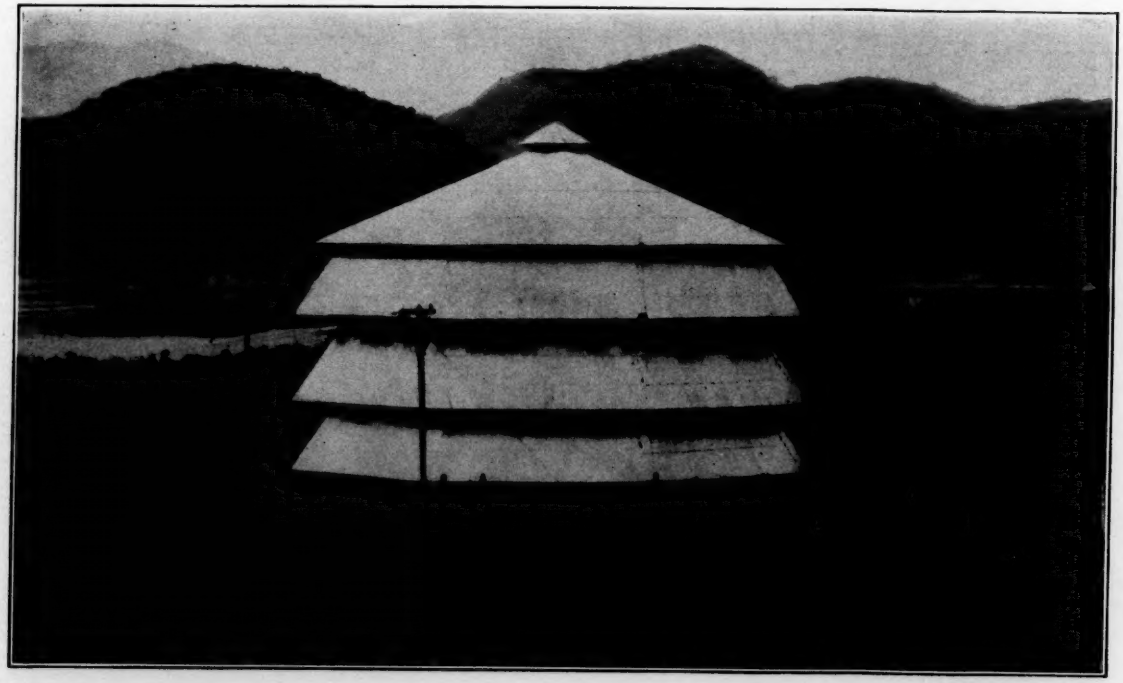
Practically all loss of gasoline from these storage tanks might be expected from "blow-offs" of vapor at time of filling the tank or from equalizations of pressures due to quick reduction of outside air temperatures and pressures as from a sudden cool shower, following high solar radiation from the tropical sunshine.

When the steel tanks were completely unprotected it was found by a test (taking the average for 30 days) that there was a loss of 25 gallons a day from loss of volatized gasoline; later this was cut down to about 20 gallons by painting the tank white and protecting it by the hood and upper "petticoat," seen in the photograph; later still, the two lower petticoats were added, cutting the actual evaporation losses to 8¼ gallons a day. It is not claimed that these figures are necessarily those that would be again derived from similar tests. The conditions of the usages of gasoline from the tanks are too variable for that. Taking the results as obtained, the following data is interesting, as to the financial wisdom of evaporation precautions:

Cost of hoods, petticoats, and painting of, all in white, $2,700.00.

Daily saving of gasoline, 16¾ gallons.

Annual saving in value of gasoline at 32 cents per gallon, $1,956.40.



BALBOA, C. Z., 55,000 BBL. GASOLINE TANK, SHOWING EVAPORATION PROTECTION BY MEANS OF PETTICOATS.

That is, the protection provided paid for itself in 1.63 years. The protection now provided these tanks is as effective as it is practical; vapor loss due to refilling of tank is not so frequent that an auxiliary receiver tank is worth while. The only other element that can be attacked, in order to lower further the vapor losses, is the daily simultaneous variation in temperatures, inside and outside the tank. A study of these was made between October, 1920, and March, 1921. Temperatures were taken graphically by thermographs under the hood but on top of the tank; inside the gasoline; inside the tank but in the vapor above the liquid; also pressures by barographs both inside the gasoline vapor and the outer air were recorded. A summary of this study during the period—which covers the most pronounced wet and dry season months of Panama annual weather—shows:

That the temperatures in the air space between the tank top and the hood were very similar to those in an ordinary weather bureau thermometer shelter; temperature conditions inside the gasoline vary but slightly whether the tank was full or partly full. But when the surface of the gasoline was lowered, the trace resembled in marked degree (but with reduced daily variations of values) the trace obtained from the air temperatures outside the tank but under the hood. This trace is of course the temperature of the gas vapor above the liquid gasoline. The weekly variations in the temperature of the gasoline did not exceed 5 degrees Fahrenheit, and the average daily fluctuations were about 2 to 2½ degrees. And the temperature of the gasoline was close to 75 degrees Fahrenheit in both the dry and wet seasons. It was also found that the pressure variations inside the tank were almost continually within the ¼-pound square inch outside pressure variation necessary to require "blow-offs" of vapor; the exceptions were when a change of gasoline content was made due to filling or drawing off from the tank. It was therefore concluded that the economic limit of saving had been made in the way of protection

from evaporation and that the present losses are due to minute losses at the laps of the plates, around rivet heads, valves and connections on tanks and connecting pipe lines, and "blow-offs" when gasoline is being pumped into the tank.

A concrete idea of what the hood and petticoats accomplished in the way of equalizing the temperatures from direct rays of the sun is shown by reference to the curves. These are plotted from an experiment made by The Panama Canal in April, 1920. Bars of steel, painted various colors, directly exposed to solar rays, had temperatures as shown; whereas a thermometer in an instrument shelter (painted white) gives very much smaller temperature variations. The gasoline tank, protected by the hood and petticoats, is a practical application of this experiment.

DAILY ANALYSIS OF ENGINEERING PERFORMANCES.

By Ensign R. D. Sweeney, U. S. Navy, Member.

Today engineer officers are occasionally discussing a daily analysis of engineering performances but in practice, with a few exceptions, such an analysis is unknown. The reason for this is perhaps a great amount of skepticism regarding its value—either skepticism or disregard. Then, too, it is widely supposed that the labor involved in making a daily analysis is far too great to warrant its use. It is the purpose of this article to show the source of losses found on board ship, to indicate in a clear way the simplicity and ease of making a Daily Analysis of Engineering Performances and to point out its value to engineers who have not come into contact with engineering on board ship.

By plant analysis is meant the comparison of the actual boiler efficiency with the maximum obtainable efficiency, and of the actual steam consumption of various auxiliaries and main engines with the designed steam consumption of such auxiliaries and main engines. By balancing the plant's theoretical or guaranteed performance against its actual performance, faulty operation can be discovered and prevented. Therein lies its value.

To effect such a balance the water rate of each steam machine while in good condition and the rate of evaporation of the boilers must be known. The water rate of each machine is best taken from the builder's curves, which give the guaranteed performance. The rate of evaporation of a boiler can be determined by making boiler tests from time to time.

Economy begins first with the boilers. Boiler losses may be enumerated as follows:

1. Loss due to moisture contained in the fuel.
2. Loss due to moisture formed in the burning of hydrogen.
3. Loss due to moisture in the air.
4. Loss due to heat carried away in dry chimney gases.
5. Loss due to the incomplete combustion of carbon.
6. Loss due to carbon appearing in unconsumed refuse.
7. Radiation losses.
8. Losses due to unburned volatile hydrocarbons.

None of the above losses can be separately calculated on destroyers, but boiler tests can be made and this, with a study of the flame and smoke will indicate whether or not the combustion has been good. Combustion may be complete but not perfect, but perfect combustion must be complete. Perfect combustion is a state in which just enough air and hence just enough oxygen enters the furnace to burn the carbon in the fuel, while in complete combustion the air may be in excess, therefore containing more than enough oxygen to burn the carbon and that which is left over cools the gases of combustion.

Since oil as a fuel has nearly replaced coal in naval vessels, everything said in this article will apply mainly to oil burning ships.

In everyday boiler operation on board ship the best practicable means of obtaining high boiler efficiency is a study of the flame and of the smoke pipe. A black smoke is an indication of not enough air, provided the burners are clean. A clear stack or white smoke indicates excess air. Three or four hundred per cent excess air is not uncommon. The invisible or practically colorless gases issuing from a stack may represent a combustion loss many times as great as that due to the carbon present in the gases when making a slight amount of smoke, and but a small amount of such carbon is sufficient

to give color to large volumes of gases. A light brown haze is the sign of the best combustion. Carbon on the atomizers indicates that they are improperly placed. A short flame full of sparks indicates excess air and dirty atomizers. The following table shows the temperature and appearance of flame:

Appearance of Flame	*Temperature Degrees Fahrenheit*
Dark Red	970
Dull Red	1295
Dull Cherry Red	1475
Full Cherry Red	1645
Clear Cherry Red	1825
Deep Orange	2000
White	2375
Bright White	2555
Dazzling White	2725

The proper control of air registers is a very important feature. The area in conical registers through which the air passes varies considerably with different makes which use the same atomizer. By closing down on the registers better atomization can be obtained. On the battleship *Arizona* during an economy run, all conditions remaining the same, by closing down on the registers, it was found that the air pressure in the fireroom could be reduced from 2.8 inches to 2 inches and 200 gallons of fuel oil saved an hour.

It follows, then, that the regulation of the air supply to the boiler is the most important factor in obtaining and maintaining combustion efficiency. If the draft is too little the percentage of CO in the gases will be large and the steam will be generated uneconomically. If there is too much draft, experience has shown time after time, that the steam will be made wastefully.

In actual practice it is necessary to introduce more air than is theoretically necessary in order to insure perfect and complete combustion. This excess air has been cut down to six per cent.

One common cause for poor boiler efficiency on board ship is that the casings are not kept tight. To have a leaky setting means fuel waste and to have a properly tight setting means attention. Air leaks in the casing chill the gases of combustion and the heating surfaces. Air leaks in the casings, uptakes and firerooms make it necessary to speed up the blowers, resulting in a greater steam consumption.

Figure 1 is a curve in which the percentage preventable fuel loss is plotted against the percentage of CO_2 in the gases of combustion, and the percentage excess air by volume and shows very clearly the loss due to excess air.

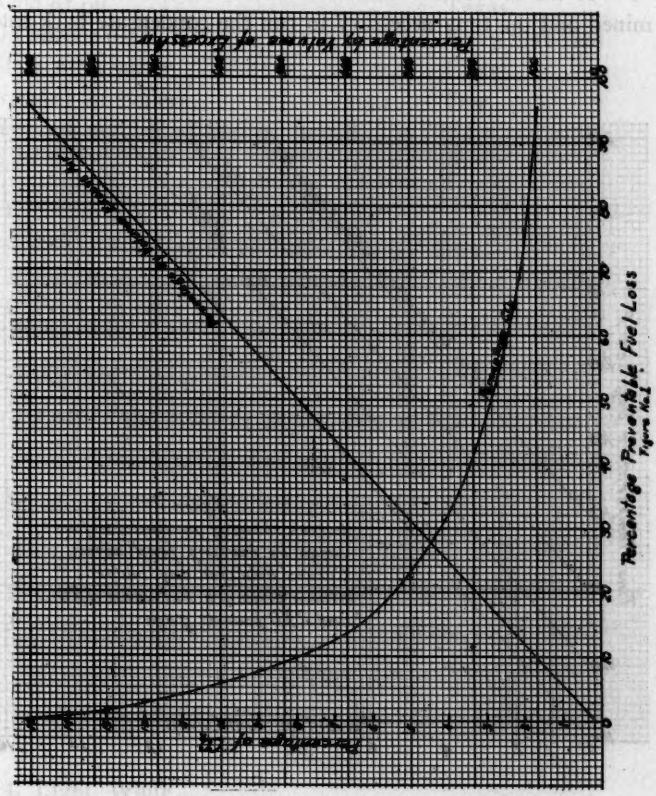
Figure 2 is a set of curves which show the percentage of fuel saved by heating boiler feed water. The curves hold for any boiler gauge pressure between 150 and 250 pounds. Each curve is for a different initial temperature of the feed water. The co-ordinates for these curves were calculated from the following formulas:

$$\text{Percentage of fuel saved} = 100 - \frac{(t_s - t_f + L_p)\ 100}{t_s - t_i + L_p}$$

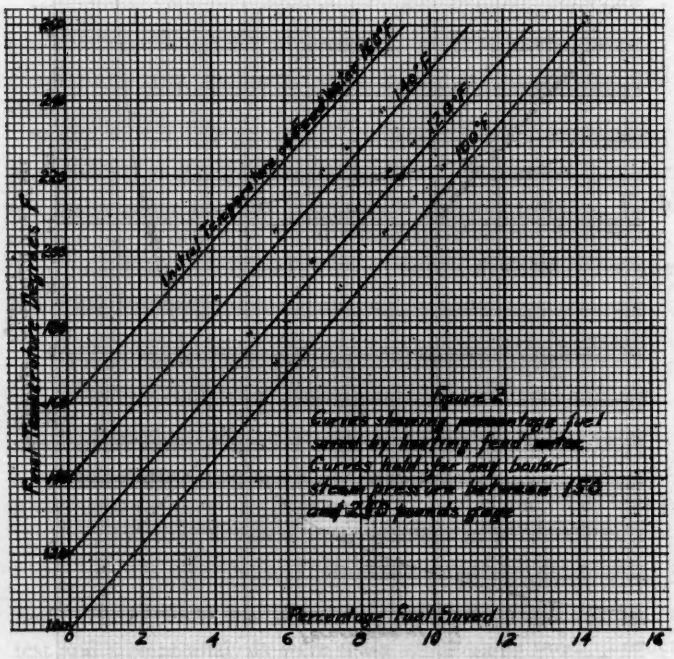
p = boiler pressure pounds by gauge.
t_s = temperature, degrees F., of steam, corresponding to p.
t_f = temperature, degrees F., of feed water.
t_i = initial temperature, degrees F., of feed water (feed tank).
L_p = Latent heat in steam corresponding to p.

The fuel consumption of the U. S. T. B. Destroyer *Stringham,* Number 83, was excessive while at anchor. A boiler test and a plant analysis were made at the same time, in order to determine the source of the losses. No special measuring devices were available. Such tests can be conducted on all destroyers and most battleships.

During the test the feed pump suction was taken from C-101, which is a feed bottom directly below the forward engine room floor plates. All drains and exhaust steam were led to the auxiliary condenser. The auxiliary air pump discharged


Percentage Preventable Fuel Loss
Figure No.1

to the feed tank and all valves in suction lines from the feed tank were secured. When the condensate in the feed tank reached 700 gallons it was run back into C-101. The rate of flow of the condensate into the feed tank was carefully determined and the time necessary to run the condensate back into



C-101 was noted and the necessary correction applied. The make-up feed was carefully measured. The height of the water in the boiler was maintained constant.

The fuel oil used was measured by an oil meter and checked with carefully taken soundings.

The following is the result of the test:

Date of test	November 3, 1921.
Duration of test, hours	Four.
Kind of oil	Texas.
Oil burner used	Navy Standard.
State of weather	Clear.
Number of boilers	One.
Average number of burners in use	2.1.

Average Pressures.

1. Steam Pressure by gauge, pounds	175
2. Draft Pressure in Fireroom, inches water	2.2
3. Oil Pressure by gauge, pounds	208
4. Feed Pressure by gauge, pounds	250

Average Temperatures.

5. Oil, degrees F	167
6. Feed Water entering Boiler, degrees F	216.2

Oil.

8. Gallons used during test	313
9. Average gallons used per hour (by meter)	78.25
10. Standard allowance per hour, gallons	45
11. Competition ratio during test	57.507
12. Weight of oil used during trial, pounds	2445.00

Steam.

13. Assumed Quality, per cent	97

Smoke.

14. Light White	
15. Factor of Evaporation	1.017
16. Factor of Correction for Quality of Steam	.9749

Water.

17. Total weight of water fed to boiler, corrected for make up feed, pounds	29,714.888

18. Equivalent weight of water evaporated into dry steam, pounds	27,968.044
19. Equivalent weight of water evaporated into dry steam from and at 212 degrees F., pounds.	28,443.50
20. Equivalent evaporation from and at 212 degrees F. for steam 97 per cent quality	30,219.95

Fuel Oil Per Hour.

21. Oil per hour, pounds	611.25
22. Square feet heating surface, 1 boiler	6885
23. Oil per hour per square foot heating surface, pounds	.0887
24. Oil per hour per burner, pounds	291.0
27. Actual evaporation per square foot heating surface, pounds water per hour	1.078
28. Equivalent evaporation per square foot heating surface from and at 212 degrees F. quality of steam, 97 per cent, pounds per hour	1.096

Economic Results.

29. Water evaporated per pound oil, pounds	12.152
30. Equivalent evaporation from and at 212 degrees	12.358

Efficiency.

31. Boiler Efficiency	63.365
32. Boiler Horsepower	217.3

In analyzing the performance of the auxiliaries of the *Stringham* the steam consumption of the pumps was taken from curves which gave the guaranteed consumption of the unit. Such a curve is shown in Figure 3. It must be remembered that the same size and make of pump operating under different conditions of discharge pressure will have different curves. A Fire and Bilge Pump will have a different consumption curve when discharging into a fire main from what

it will have when pumping bilges. The steam consumption of the evaporators and generators was determined as explained in the following paragraphs.

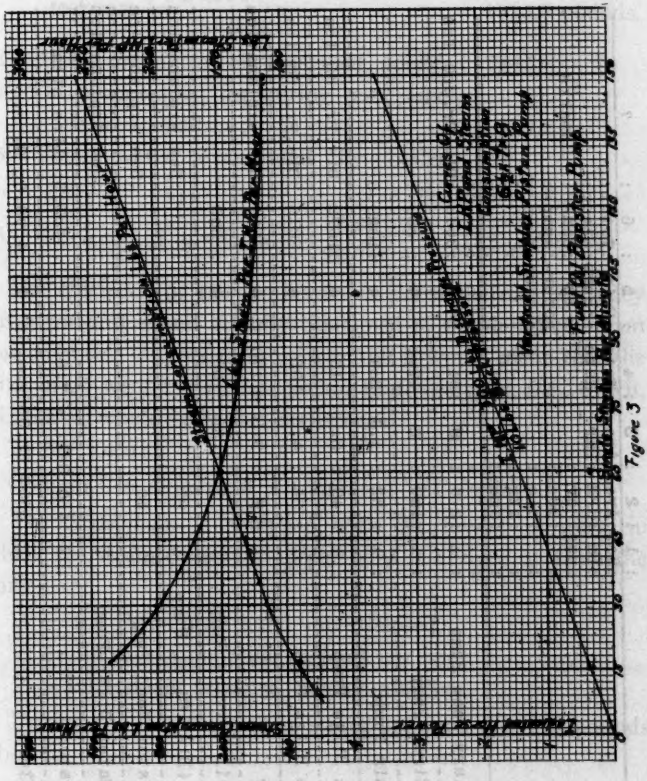


Figure 3

All data necessary to make a plant analysis, with the exception of the boiler test, can be taken from the engine room log if it is properly and carefully kept. The main factors are discharge pressure, R.P.M. or strokes per minute, gallons of water distilled and K.W. hours generated. If underway, the shaft horsepower, steam pressure in high pressure steam chest and vacuum must be known.

The following form shows the results of the plant analysis on the *Stringham*:

	1	2	3	4	5	6	7
	No. in use	Ave. RPM or Double stroke	Ave. IHP	Water Rate	Unit hours in operation	Total steam Consumption	Remarks
Auxiliary Feed Pump	1	2.8	8	84	4	2688	Dis.Pr.250 lbs
Aux. Circ. Pump	1	150	2	47.5	4	380	
Aux. Air Pump	1	32	1.1	234	4	1029.6	
Fire & Bilge Pump	1	40	3.8	110	4	1672	Dis.Pr. 36 lbs
Fire & Bilge Pump	1	36	4.8	105	4	2016	Dis.Pr. 50 lbs
Blower	1	#	9.8 x 170 x		4	6664	See explanation
Fuel Oil Service Pump	1	9	1.25	160	4	800	
F & B (Dist.Circ.)Pump	1	80					See explanation
Evaporator Feed Pump	1	40					See explanation
Fresh Water Distr.Pump	1						See explanation
Evaporators	DE	450 Gal. x 7.63				3433.5	Live steam
Generators	1	3600	21.056x54.6x		4	4598.6	G. E.

(1) Total (Guaranteed Steam Consumption) 23,281.7

(2) Equivalent amount of oil at boiler test rate $= \frac{23{,}281.7}{12.152} =$ ..... 1915.8 pounds.

(3) Actual amount of oil consumed, pounds 2445.0

(4) Oil unaccounted for during test (auxiliary) pounds..... 529.2

(5) Total water evaporated by boiler, pounds 29,714.8

(6) Efficiency of auxiliaries $= \frac{23{,}281.7}{29{,}714.8} = .783$ or 78.3%.

The efficiency of the auxiliaries is not a thermal efficiency but a ratio of steam actually consumed to the designed steam consumption. If this ratio is over 100 it would mean that the auxiliaries were taking less steam than designed for, which, of course, would be a rare occurrence.

In the form:

Column 6 = Column 3 × Column 4 × Column 5.

The *Stringham* is equipped with Sturtevant Blowers, in which the steam consumption per hour is arrived at from the formula:

$W = 9.8\ P$, with all hand valves closed.

W = Steam consumption, pounds per hour.

P = Inlet steam pressure, pounds absolute.

During the test the blower steam pressure was 170 pounds absolute.

Also,

$W_1 = 14.2\ P$, with one hand valve closed.

$W_2 = 18.8\ P$, with two hand valves closed.

The constant appearing in the above formula depends upon the nozzle, and applies to Sturtevant Blowers of the following type: VD-5, 27,000 cubic feet capacity, 7-inch air pressure, 1350 R.P.M.

The following formulas give the steam consumption of Terry forced draft blowers, Type G. V., 30,000 cubic feet capacity, 6 inches air pressure, 1470 R.P.M.:

W = 33 P with all hand valves closed.
W = 41.5 P with one hand valve open.
W = 50.5 P with two hand valves open.
W = 59 P. with three hand valves open.
W = 67.5 P with four hand valves open.

Recently conducted tests show that it takes .07 gallons of fuel oil to distill one gallon of water at a boiler rate of evaporation of 14 pounds of water to a pound of oil, the evaporators being operated in double effect with a first effect coil pressure of 75 pounds gauge. This .07 gallons of oil not only included the oil necessary to generate steam for the evaporator coils but it also includes the oil necessary to generate steam to operate the distiller circulating, evaporator feed, fresh water, vacuum and brine pumps and to blow down the evaporators every four hours. Hence the steam consumption obtained by multiplying the gallons of water distilled by the pounds of steam necessary to distill one gallon will include the steam consumption of the evaporator auxiliaries. However, if the water is distilled using exhaust steam, in making a plant analysis, the consumption of each auxiliary of the evaporator plant must be computed separately and the gallons of water distilled must not be taken into consideration.

.07 gallon fuel oil equals 1 gallon water distilled.

.545 pound fuel oil equals 1 gallon water distilled.

1 pound fuel oil equals 14 pounds water evaporated by boiler.

.545 pound fuel oil equals 7.630 pounds water evaporated by boiler.

Hence, to distill one gallon fresh water it will require 7.630 pounds of steam.

The *Stringham* is equipped with two 25 K. W. General Electric Turbo-Generator sets. During the test the average electric load was 21.056 K.W. The generator exhausted into the auxiliary condenser against 5-inch vacuum. The water rate for this generator with this load and vacuum is 54.6 pounds of steam per K.W. hour. The horsepower developed by the turbine was 30.351.

On battleships fitted with 300 K.W. turbo-generator sets with independent condensing plants the water rate for full load including all the dynamo auxiliaries is 35.5 pounds of steam per K.W. hour with 27 inches vacuum in the dynamo condenser. One set is not sufficient to carry the load, which necessitates the use of two sets at a little over half load, increasing the water rate to 44 pounds of steam.

After a boiler test and plant analysis have been made the results will show one of the following cases: Case 1—High boiler efficiency and high efficiency of the auxiliaries; Case 2—Low boiler efficiency and low efficiency of the auxiliaries; Case 3—Low boiler efficiency and high efficiency of the auxiliaries; Case 4—High boiler efficiency and low efficiency of the auxiliaries.

Case 1, of course, will give the lowest oil consumption and is the condition to be attained. Case 2 will give the greatest oil consumption. If the results obtained are classed under Case 2, immediate action must be taken to increase the boiler efficiency and to rectify the operation and condition of the auxiliaries. In Case 3 the cause (there is always a cause) of the low boiler efficiency must be determined and rectified. In Case 4 the auxiliaries are not being properly operated. If nothing is visibly wrong with any of the auxiliaries and their operation, separate steam consumption tests should be run on each unit until the unit or units that are increasing the designed steam consumption of the plant has been determined.

There should be no stopping off place in economy, but the fact that each vessel has a set of allowances has provided a

stopping off place. When a performance ratio of 100 has been obtained a few perhaps think that further improvement is not possible. Tests and analysis prove the fallacy of this.

When making tests and analysis each hour should be analyzed separately. Calculate the economic results of the boiler test and plant analysis each hour. Then compare the hours. If one hour shows a larger oil consumption than another the boiler test and analysis will show why. Plant analysis and boiler test are tell-tales that indicate that something is wrong, and by studying their results the defects can be determined and then remedied.

The results of the boiler test and plant analysis on the *Stringham* come under Case 2—low boiler efficiency and low efficiency of the auxiliaries. In determining the source of losses start first with the boiler. The boiler efficiency was 63.365 per cent and the rate of evaporation was 12.152 pounds of water per pound of oil. The average number of burners in use was 2.1, the average air pressure 2.2 inches of water, the blower steam pressure was 170 pounds absolute, a light white smoke was emitted from the stack nearly continuously, the boiler casing showed air leaks and the boiler gauge glass was seven-eighths full. The flame was short and full of sparks and the burner plugs and tips were in poor condition and carbonized rapidly.

The immediate conclusion is that the air supply was too great and the atomization poor, as indicated by the smoke, study of the flame, air pressure, and the boiler test results. To remedy this defect the leaks in the casing must be stopped and of course the air pressure cut down. At the end of the test the air pressure was cut down to .9 inches, to do which only 70 pounds absolute steam pressure was necessary on the blower. This is still a high air and steam pressure, but the worn condition of the plugs and tips and the necessity of a clear stack make a slight excessive air pressure necessary. The

blower speed and volume of air output vary almost in a direct ratio to the blower steam pressure or to the manometer readings. Then $\frac{170}{70} = 2.42$ or 242 per cent; or $\frac{2.2}{.9} = 2.44$ or 244 per cent. This ratio is not exact but it is safe to say that the excess air was 200 per cent, which is not uncommon. Referring to the curve in Figure No. 1, with an excess air supply of 200 per cent, the preventable fuel loss is 20.6 per cent.

Total oil consumed in 4 hours, pounds.....2445

(*a*) $2445 \times .206 = 503.57$ pounds oil, preventable loss.

The back pressure was 6 pounds, which should have heated the feed water to at least 224 degrees F. The average temperature of the feed water was 216 degrees F., due to the fact that the feed heater drains were not properly operated at times. The difference between 216 degrees F. and 224 degrees F. temperature of the feed water created a loss of 1 per cent in the total fuel burned.

(*b*) $.01 \times 2445 = 24.45$ pounds oil loss.

The loss due to air leaks cannot be easily calculated. The water level in the boiler was carried too high and the feeding of the boiler was not steady, which gave a poor quality of steam. As the quality of steam cannot be determined on a destroyer, this loss cannot be calculated exactly, but is taken care of when calculating the loss in the auxiliaries, as the designed steam consumptions are calculated for dry steam.

The carbon on the atomizer tips was an indication that they were not properly located. Flexible leads on the atomizers permit of adjustment in and out. The angle of spray varies for the same size plug and tip even if made by the same machinist. An atomizer is properly located when the flame just clears the cone brick. All the oil that strikes a cone brick is a loss. When a burner is not in use it should be withdrawn as far as possible.

Very often in installing new cone bricks, care is not taken to see that the cone brick makes a smooth contact with the conical register, and a continuation of the perfect cone of the register. Not enough attention is given to keeping the registers clean and tight when closed and to the cleaning of plugs and tips, which have often been damaged due to cleaning with sharp, hard instruments or emery paper.

There is a considerable disregard of the proper heating of oil. If no viscosimeter is available, the flash point and gravity Baumé can always be obtained from the tanker or fueling station. There is a set of temperature-viscosity curves in a pamphlet published by the Bureau of Engineering, and with a flash point and viscosity as arguments, a curve can be found which will fit the oil in question. At a point on this curve where the viscosity is between two and four Engler, note the temperature. This will be the proper temperature to which the oil should be heated. If this is above the flash point the oil should be heated to its flash point. These are a few primary principles of fuel oil burning known perhaps by all but observed by few.

The total preventable fuel oil loss in the boiler that could be easily calculated on the *Stringham* was 528.02 pounds. In calculating the loss the remedy is plainly shown.

The next step is to consider the auxiliaries. The loss due to improper speed and operation of each unit can be calculated. The analysis shows a loss of 529.2 pounds of oil in four hours or 3175.2 pounds a day. This does not indicate by any means that only 529.2 pounds could be saved during the test. Assuming that the auxiliaries were operating at their proper speeds, it does show that the plant was wasting 21.7 per cent of the steam generated by the boiler.

(*c*) Loss due to inefficiency of auxiliaries = 529.2 pounds oil.

FORCED DRAFT BLOWER.

Cutting down the air pressure in the fireroom not only increases the boiler efficiency but cuts down the blower steam consumption.

Blower pressure during test.....	170 pounds absolute
Necessary blower pressure.....	70 pounds absolute
Actual steam consumption $= 9.8 \times 170 \times 4$ (hrs.) $=$	6664 pounds
Necessary steam consumption $= 9.8 \times 70 \times 4$ (hrs.) $=$	2744 pounds
Steam loss due to excessive speed $=$	3920 pounds

(*d*) Oil loss at boiler test rate of evaporation $= \frac{3920}{12.152}$ $= 322.5$ pounds.

Blowers generally have four hand nozzles and a fifth nozzle which is always open. In port, on destroyers and most battleships, it is not necessary to open any of the hand nozzles, but often the hand nozzles are opened when it is not at all necessary. This increases the steam consumption.

Electric blowers have been used on a few destroyers in port. This blower is placed in the escape hatch or hung in front of the registers in use, and the fireroom left open. While the fuel consumption is decreased, there is considerable danger of a flare back, the tubes become dirty sooner, and less water can be distilled in the evaporators using auxiliary exhaust. The overall efficiency of the plant is but a little increased.

GENERATORS.

The turbo-generator sets on the *Stringham* are General Electric, 25 K.W. The vacuum in the condenser was 5 inches. With this vacuum each K.W. hour generated, cost 54.6 pounds of steam. Referring to the steam consumption curves, it is seen that with the same load if the vacuum had been 25 inches, each K.W. hour would have cost 48 pounds of steam.

Actual steam consumption — 21.056 × 54.6 × 4 (hrs.) = ... 4598.6 pounds
Steam consumption, 25″ vacuum — 21.056 × 48 × 4 (hrs.) = 4042.7 pounds

Steam loss due to low vacuum = ... 555.9 pounds

(*e*) Oil loss at boiler rate test of evaporation. 45.7 pounds

GENERATOR DATA.

G. E. Generator (Builders test rates).
3600 R.P.M. 0 degrees superheat.
200 pounds Gauge Steam Pressure.

Non-condensing.

Load	*Exh. Pres.*	*W.R. lbs. per K.W. Hr.*
1/2	0	68
Full	0	51.5
5/4	0	48.5
1/2	10#G	88.5
Full	10#G	68
5/4	10#G	63.5

Condensing.

Load	*Vacuum, Inches*	*W.R. lbs. per K.W. Hr.*
1/2	25	60.5
Full	25	47
5/4	25	45.5
1/2	28	57
Full	28	44.5
5/4	28	43

Approximate formulas for computing pounds of steam per hour for G. E. Generators:

Exhaust Pressure 0 pounds gauge: W = 410 — 35 K.W.
Exhaust Pressure 10 pounds gauge: W = 500 — 48 K.W.
25 inches vacuum W = 320 — 34 K.W.
28 inches vacuum W = 300 — 32 K.W.

Westinghouse Generator Sets.

Load	*Exh. Pres.*	*W.R. lbs. per K.W. Hr.*
1/2	10#G	117
Full	10#G	74

Knowing the water rate of the generator and the rate of evaporation of the boiler, the cost of running searchlights, electric blowers, etc., can be computed. For example, say the ice machine motor takes 10 ampères, 125 volts. The generator is G.E. make and the average load is 50 per cent with 0 pounds gauge exhaust pressure. At this load and exhaust pressure each K.W. hour takes 68 pounds of steam. Suppose the boiler evaporates 15 pounds of water per pound of fuel oil,

Then: $\frac{10 \times 125 \times 68}{1000 \times 15} = 5.67$ pounds of fuel oil per hour (ice machine).

PUMPS.

Pumps are large consumers of steam. Steam cannot be used expansively in direct acting pumps because the fluid pressure in the barrel is practically constant, hence the steam pressure at all parts of the stroke must be sufficiently high to overcome the fluid pressure. If the steam were used expansively the pressure would be too high at the beginning and too low at the end of the stroke. It is vitally important that pumps should be run at the slowest speed possible with full length stroke.

Various boiler tests have given low efficiencies of water ends of feed pumps. One reason for this is the fact that suction must be taken from C-101 during the test and hence there is a suction lift. At the end of each stroke the water in the suction line runs back into C-101 and part of the next stroke is wasted by lifting the water back up to the pump. Water above 160 degrees F. must come to the pump under a head and this head increase until a 12-foot head is necessary at 212 degrees

F. In taking suction from C-101, the pump must not only be run faster but the heat of the water in the feed tank is lost. In taking suction from the feed tank the water will come to the pump under a head and in this case many tests have given an efficiency of the water end of 70 per cent.

Efficiency of water end of pump (suction C-101) = ... 12.6 per cent.
Efficiency of water end of pump, with head = ... 70 per cent.
Single strokes actually made (12.6 per cent efficiency) = ... 10 per minute.
Single strokes actually made (70 per cent efficiency) = ... 2 per minute.
Steam consumption per hour, pounds = ... 1150 at 12.6 per cent efficiency.
Steam consumption per hour, pounds = ... 550 at 70 per cent efficiency.

Steam saved per hour, pounds..... 600

Oil saved per hour (rate of evaporation = 14 to 1) $\frac{600}{14}$ = 43 pounds.

43 pounds = 5.5 gallons.

Hence the oil saved by one pump by taking suction from the feed tank is 5.5 gallons per hour or 132 gallons per day, provided that the pump is operating at its best efficiency.

The errors in operation of pumps may be stated as follows: shortened stroke, over-speeding, piston rings too tight and pumps out of line, causing scoring of the liners, piston rings leaky or broken, pounding, leaky valves, and leaky stuffing boxes.

DISTILLING PLANT.

Very great progress has been made in the operation of distilling plants in the last year, especially in the Destroyer Squadrons, Atlantic Fleet. It is safe to say that any destroyer in these Squadrons is self-supporting in water when acting singly, using auxiliary exhaust steam in the coils. A vacuum is maintained on the shells by means of the distributing pump or by installing a small line from the auxiliary and main condensers to the measuring tank, which is made air tight. This has resulted in maintaining a vacuum of twenty-one inches in the shells, and temperature differences as high as 74 degrees F.

without priming. The density of the water in the shells must be kept as low as possible and never above three-thirty seconds.

Only 450 gallons of usable water was distilled on the *Stringham* during the four hours of the test, using live steam double effect. It was necessary to throw away 100 gallons of water distilled, due to the fact that the salinity was too great for use. The capacity of the *Stringham's* evaporators is 7500 gallons per day in double effect. The steam pressure gauge was 75 pounds, the first effect shell pressure 15 pounds and the second effect shell 4 pounds gauge. One reason for the low output was the fact that the coils were very dirty, as a 75-pound coil pressure gauge should give a 35-pound pressure gauge in the shell with the vapor valves to the second effect coil wide open. Other reasons for low output was that no attention was paid to the density of the brine except by the procedure of blowing down each watch, which is not entirely correct and because the feed water was not heated and entered the shells at 70 degrees F. instead of 180 degrees F., which is the average temperature for evaporator feed water. This caused a 10 per cent heat loss in shells alone.

The distiller circulating pump was run too fast. The loss due to this pump is included in the total loss of the evaporating plant. The temperature of the water in the measuring tank should govern the rate of speed of the distiller circulating pump.

The coil drains were properly operated and no steam was blowing through. The coils were kept free of air. The loss due to the necessity of throwing away 100 gallons of water is equal to 100 times 7.63 divided by 12.152, which equals 62.7 pounds of oil.

(f) Loss due to evaporating plant = 62.7 pounds oil.

Adding the oil losses (a), (b), (c), (d), (e) and (f) will give a total loss of 1488.12 pounds oil during the four hour test or 190 gallons. The total oil consumed in the four hours

was 313 gallons. 313 — 190 = 123 gallons for four hours, or 738 gallons per day. Many destroyers consume less oil than this per day at anchor.

If underway, a plant analysis can also be made. The steam consumption of the various auxiliaries can be obtained as described above. When dealing with turbines, the action of the steam in the turbine must be understood. Briefly, steam flow through turbines follows the laws governing adiabatic expansion, as in the Rankine Cycle. In adiabatic expansion the steam neither receives heat from any external source nor gives out heat externally, and all work done in the turbine is at the expense of the internal heat of the steam, the pressure of the steam falling and the quality becoming poorer. As steam flows through a nozzle or reaction blade it expands continually and each particle of steam as it expands pushes the particle ahead of it forward at a faster rate, and so increases the velocity of flow. Since in adiabatic expansion no heat is added to or subtracted from the steam, the entropy is constant. This is not exactly true, as there is a reheat due to blade friction, which will cause an increase in entropy. However, in practice steam is always assumed to expand adiabatically at constant entropy. Expansion at constant entropy is also known as isentropic expansion, which must not be confused with isothermal expansion.

There are many formulas for calculating the internal heat or energy of the steam, but it is obtained very simply from Mollier's Diagram, as shown in the following problem:

In making a plant analysis underway the water rate of the main turbines can be calculated as follows. Suppose the latest standardization curves show a shaft horsepower of 3540 at 20 knots and that the mechanical efficiency of the reduction gear, bearings, etc., is 90 per cent, which repeated tests have proven to be the general efficiency under the conditions assumed (this is not the gear efficiency proper):

$$\text{H.P.} = \frac{\text{S.H.P.}}{\text{mechanical efficiency}} = \frac{3540}{.90} = 3933.$$

Pressure of steam entering turbine, pounds = 120 (absolute).

Vacuum (condenser) = 28.5 inches or .696 pounds.

Quality of steam = 98 per cent.

As steam in a turbine expands adiabatically, from Mollier's Diagram we find that at 120 pounds pressure absolute, 98 per cent quality, the total heat of the steam is 1172 B.T.U.'s, and following down the line of constant entropy to a pressure of .696 pound the total heat of the steam is 856 B.T.U.'s.

1172 — 856 = 316 B.T.U.'s used by the turbine per pound of steam. 1 B.T.U. = 777.5 foot pounds.

$$\text{H.P. (turbine)} = \frac{\text{Heat drop} \times \text{lbs. steam per min.} \times 777.5}{33{,}000}$$

$$\text{Pounds steam per min.} = \frac{33{,}000 \times \text{H.P.}}{\text{Heat drop} \times 777.5}$$

$$\text{Pounds steam per min.} = \frac{33{,}000 \times 3933}{316 \times 777.5} = 528.2$$

Pounds steam per hour = 31,692.

Pounds steam per H.P. hour = 8.058.

Pounds steam per S.H.P. hour = 8.952.

This is the theoretical water rate. Turbine efficiencies run around 60 per cent to 70 per cent. This efficiency, however, is not a thermal efficiency but is the percentage of heat drop actually converted into useful energy. Considering this factor:

$$\text{H.P. (turbine)} = \frac{\text{Heat drop} \times \text{lbs. steam per min.} \times 777.5 \times \text{Eff.}}{33{,}000}$$

$$\text{Pounds steam used per min.} = \frac{33{,}000 \times \text{H.P.}}{\text{Heat drop} \times 777.5 \times \text{Efficiency.}}$$

In the case of the stated problem, assuming an efficiency of 60 per cent, the steam consumption pounds per hour would be 52,820 and the water rate would be 13.43 pounds of steam per H.P. or 14.92 per S.H.P. steam consumption so calculated should agree very nearly with the builders water rates.

The fuel oil necessary to furnish steam for the turbines alone can be calculated as follows:

H.P. = 3933.

W.R. = 13.43 pounds per H.P.

Steam consumption = 52,820 pounds per hour.

Pounds of water evaporated per pound, oil = 15.3.

$$\text{Pounds of fuel oil necessary per hour} = \frac{52{,}820}{15.3} = 3452.3.$$

Gallons oil necessary per hour = 449.

Pounds oil per S.H.P. per hour = .975.

Suppose that in the above problem the gland steam pressure fails, causing the vacuum to fall to 24 inches. Referring to Mollier's Diagram in this case, the turbines theoretically utilize but 244 B.T.U.'s from each pound of steam, as compared with 316 B.T.U.'s utilized with 28.5 inches vacuum.

316 — 244 = 72 B.T.U.'s lost to condenser per lb. steam.

$$\frac{72}{316} = .227 \text{ or } 22.7 \text{ per cent heat loss.}$$

$$\text{Pounds of steam per min.} = \frac{33{,}000 \times 3933}{244 \times 777.5 \times .60} = 1140.2.$$

Pounds of steam per hour = 68,412.

$$\text{Pounds of oil necessary per hour} = \frac{68{,}412}{15.3} = 4471.$$

Gallons oil necessary per hour = 584 (24 inches vacuum).

Gallons oil necessary per hour = 449 (28.5 inches vacuum).

584 — 449 = 135 gallons oil loss per hour.

Or to make the same speed with a 24-inch vacuum instead of a 28.5-inch vacuum it will take 135 gallons more oil per hour or 22.7 per cent more heat must be added, which is accomplished in actual practice by raising the steam pressure in the high pressure steam chest when the vacuum falls.

In working the above problem, using 28.5 inches vacuum, the steam pressure, specific gravity of the fuel oil, shaft horsepower, vacuum, and boiler rate of evaporation were taken from the four hour official trial at 20 knots of the *Pruitt*. The assumptions were: 90 per cent efficiency of reduction gears and bearings, etc., 98 per cent quality of the steam, and 60 per cent turbine efficiency. The steam consumption and water rates were computed with no regard whatsoever of the actual results of the trial of the *Pruitt*.

The actual results of the trial were:

Lbs. water per hour, main engines	53,224.73
(Computed steam consumption)	52,820
Lbs. water per S.H.P.	15.03
(Computed W. R.	14.92)

It is seen by comparing the actual steam consumption with the computed consumption that the water rates and steam consumptions of the main turbines can be very accurately computed, even if certain experimentally proved assumptions are made.

The theoretical steam consumption per hour was shown to be 31,692 pounds. The trial of the *Pruitt* gave the actual consumption as 53,224.73 pounds.

$$\frac{31,692}{53,224.73} = .595 \text{ or } 59.5 \text{ per cent turbine efficiency.}$$

It is of interest to note that during this four-hour trial the auxiliaries consumed 5740.42 pounds of steam, hence the total steam consumption was 58,965.15 pounds or the auxiliaries at 20 knots consume about one-tenth of the total steam and the turbines about nine-tenths.

It is of value to be able to compute the turbine consumption. The consumption underway of the auxiliaries can be computed in the same manner as that used for the *Stringham*. The analysis of the auxiliaries is greatly assisted by using the "at

anchor" and "standing by" analysis, which will give the "efficiency" of the auxiliaries. Dividing the guaranteed consumption of the auxiliaries by their efficiency will give their actual consumption and add to this the computed consumption of the main turbines. Subtract this from the total actual steam consumption of the plant and the result will be the loss in the turbines. The total actual steam consumption of the plant is arrived at by multiplying the pounds of oil burned by the boiler test rate of evaporation.

The inspection of the engineering departments of one hundred and ten destroyers and several battleships within the last ten months plainly showed that the error or defects of operation as pointed out in this article existed in nearly every instance as well as a great lack of knowledge regarding water rates and efficiencies. Poor economy and defective operation are easily eliminated by the simply made boiler tests and plant analysis.

NOTES.

THE CRUISING ENDURANCE OF SUBMARINES.

Among the popular misconceptions with regard to naval material that prevailed before the war, none was more deeply rooted than the notion that a submarine boat could remain at sea only for a few days at a time, and therefore could not be employed in any operation which involved a prolonged absence from its base. This was one of the principal arguments with which Sir Percy Scott's critics sought to refute the sweeping claims he had made on behalf of the submarine only a few weeks before the outbreak of the Great War. It was put forward, not merely by civilian writers, but by several naval officers of high rank, and in view of this authoritative opinion it is not surprising that the general public came to regard the submarine as being useful only for coastal operations. Events soon showed, however, that the sea-going range of these vessels had been grossly underestimated. In the first month of the war German U boats were met with several hundred miles from their nearest base; early in 1915 they were encountered off the coast of Ireland, and by the spring of that year one boat had arrived at Constantinople after a voyage of 3450 nautical miles from Wilhelmshaven. Meanwhile small British submarines built in Canada had crossed the Atlantic under their own power, and the E 11, of 675 tons, had made a trip in the Sea of Marmora—that is, in enemy waters —lasting thirty-one days. Then, a year later, came the cruise of the German U 53 to the United States coast and back, a voyage of at least 7000 miles, which was accomplished without replenishing the supply of fuel. It may be said, therefore, that the cruising endurance of submarines proved to be about ten times greater than pre-war forecasts had indicated. There is still some uncertainty as to the maximum period of time which a submarine spent continuously at sea during the war. The British boats on the Atlantic patrol frequently made voyages of well over one month's duration, while according to German accounts the first *Untersee-Kreuzer*, the U 139, made one cruise lasting six weeks.

Interest in this question of submarine endurance has now been revived by the remarkable performance of a French boat, the *Victor Réveille*, which was recently ordered to carry out a test for determining how long she could remain at sea. Leaving Cherbourg at the beginning of July, she arrived at Toulon on August 31, after an uninterrupted voyage of sixty days. This, so far as is known, constitutes the longest voyage which has been made by a submarine up to the present date, though it need hardly be said that peace conditions are far more favorable for such a test than those which obtain in war. The *Victor Réveille* was formerly the U 79, and is one of the many ex-German submarines now under the French flag. Completed at Hamburg in 1916, she belonged to a group of ten boats which were specially designed for mine-laying, and one of which sowed the mines that sank H. M. S. *Hampshire*, with Lord Kitchener on board. The performance of the *Victor Réveille* is the more noteworthy in view of her

modest dimensions. On the surface she displaces only 755 tons, increased to 833 tons when submerged. Her length overall is 186 feet 4 inches, the breadth 19 feet 10 inches, and the draught 15 feet 1 inch. She is therefore less than two-thirds as long as a modern destroyer and some 10 feet narrower in beam. The propelling machinery consists of two sets of Diesel four-cycle six-cylinder engines, developing 900 brake horsepower, which gives the boat a surface speed of 10.6 knots. For submerged propulsion the electric motors develop 800 brake horsepower for 8 knots. In his paper on German submarines, Mr. A. W. Johns, R. C. N. C., said that the class to which this boat belongs differs from all the ordinary U boats in having internal main ballast tanks. They have a pressure hull of large diameter—16.4 feet, as against 13.5 feet in U 161—and small external saddle tanks, which, instead of carrying water ballast, carry oil fuel. The two main ballast tanks are internal, the larger extending under the mine room and up the sides of the motor room, while the other is under the engine-room and partly up the sides of that room. With a full load of fuel, amounting to 87 tons, the nominal range of the *Victor Réveille* at an economical speed of 7 knots is 7800 miles. It is not yet known what distance she actually covered in the course of her sixty days' voyage, but it is evident that had a speed of 7 knots been maintained throughout, the total run would be 10,080 miles, or 2280 more than she is supposed to be capable of making on one load of fuel. In point of fact, the radii of all the German boats with which our experts have had experience have proved to be substantially larger than the official German estimates, and there is more than a suspicion that their actual endurance was purposely minimized for obvious military reasons. That the submarine possesses a much wider radius of action than any surface warship of equivalent size is now well known, but it is doubtful whether the full extent of her advantage in this respect is commonly realized. She derives it mainly, of course, from her internal combustion engines, which at low power are very economical in fuel consumption, while the structural arrangement of the hull is well adapted to the carriage of a large supply of oil in proportion to the displacement. Official data published since the war show that great cruising endurance was aimed at in the design of all German submarine types. In the U 29, a typical U boat of 675 tons displacement, the endurance at an economical speed of 8 knots was 9800 miles. In the U 43, of 725 tons, it increased to 11,250; in the *Untersee-Kreuzer* Nos. 139 and 142 it was respectively 18,000 miles at 8 knots and 20,000 at 6 knots. Still more remarkable were the corresponding figures for the smaller types; the UB 18, of only 272 tons, was credited with an endurance of 7000 miles at 5 knots; the UB 48, of 516 tons, with 9000 miles at 6 knots; and the UC 16, of 410 tons, with 8700 miles at 7 knots. These are boats whose limited dimensions would seem to put them in the category of coastal submarines, yet they were nevertheless capable of voyaging for many thousands of miles. The longest-range submarines built in Germany appear to have been the *Deutschland* and her six sister boats, originally designed as submersible cargo ships, but afterwards converted into fighting craft. Although of only 1510 tons displacement, their fuel supply was sufficient for a continuous run of 25,000 miles at a speed of 5½ knots. However, when estimating the sea endurance of small vessels, whether of the surface or submersible type, it is necessary to take other factors besides the mechanical into consideration.

That a boat such as the UB 18 could really make an uninterrupted voyage of 7000 miles may well be doubted. Owing to the cramped accommodation below and above deck, the difficulty of ventilating the narror interior spaces packed with machinery, the noise of the Diesel engines, and the violent

motion of the boat in rough weather, the health of the crew would inevitably begin to suffer after a few weeks at sea. In other words, the human element would be liable to become exhausted long before the mechanical element had reached its limit. In time of war the conditions of service on board would, of course, be infinitely more trying; there would be a constant strain on the nerves of the men when they were at sea, especially in enemy waters, and therefore had to spend most of their time under hatches. The actual radius of any type of submarine in war time would therefore depend primarily on the mental and physical stamina of the complement, which implies that it cannot be fixed with any degree of precision. But as submarines increase in size, enabling better living quarters to be provided and setting free more space on deck for exercise and recreation, the conditions of service tend to become more tolerable, and there is consequently less likelihood of a breakdown occurring in the health or *morale* of the *personnel*. There is no visible reason why a very large ocean-going submarine should not be able to remain at sea as long as her fuel holds out, without imposing any superhuman strain on the *personnel*. On the other hand, if dimensions are increased beyond a certain limit the submarine becomes difficult to manœuver below the surface and requires to be navigated with extreme care in all but the deepest waters. For this reason every navy will no doubt continue to build boats of moderate size—say, up to 1000 tons—for general service, while at the same time developing the large submersible cruiser type for special overseas operations. This division of types seems, in fact, to have been adopted in the United States, France and Japan, following the example set by Germany during the war. Although the prospect of a long period of peace has naturally slowed down the development of the submarine boat, steady progress is still being maintained, and there are boats now under construction abroad which appear to be little, if at all, inferior to the German submersible cruisers in respect of sea endurance. The advent of submarines capable of crossing and recrossing the Pacific Ocean with a margin of 10,000 miles to spare promises to introduce a new and complicating factor into naval strategy.—"The Engineer," Sept. 15, 1922.

EXHAUST VALVES OF NICKEL-CHROMIUM ALLOY.

"NICHROME," AN ALLOY PRODUCED BY DRIVER-HARRIS COMPANY, SAID TO BE PRACTICALLY IMMUNE TO PITTING, WARPING AND THE OTHER DESTRUCTIVE FORCES.

Production of a poppet-valve metal, or alloy, that would better withstand the destructive forces that are created in the operations of high as well as low-powered internal-combustion motors has been a problem for many years, and the solution to which should be welcomed by every designing engineer and oil-engine builder. Because of the severe conditions under which big exhaust-valves of high-powered Diesel engines—some up to ten inches diameter—have to work it is only natural that metallurgists have made considerable efforts to develop on a commercial scale steels and alloys that would overcome the need for constant regrinding and replacing of valves made necessary by the deteriorating effects of heat, the products of combustion, and heavy mechanical stresses together with outside influences such as high-sulphur oils, salt water, etc. While certain progress has been made from time to time, the results attained have invariably betrayed weaknesses in one way or another, that little was actually gained over the orig-

inal and ordinary cast-iron and steel valve which has a tendency to carbonize, warp and deteriorate. Any metal subject to such, naturally has a possibility of producing trouble at unexpected moments, so that the operator cannot place thorough reliance upon that part of the engine under service conditions.

One of the arguments raised against the four-cycle type Diesel-oil engine for marine work by two-cycle engine-builders is that, when using heavy crude and residual oils such as Mexican fuel—particularly oil with sulphur varying from 1 to 3 per cent, additional work is given to the engine-room crew owing to the exhaust-valves becoming pitted and requiring regrinding every voyage, and sometimes twice per voyage. This, of course, does not occur so often when burning Diesel-oils, or heavy crudes without sulphur contents such as Taraken oil.

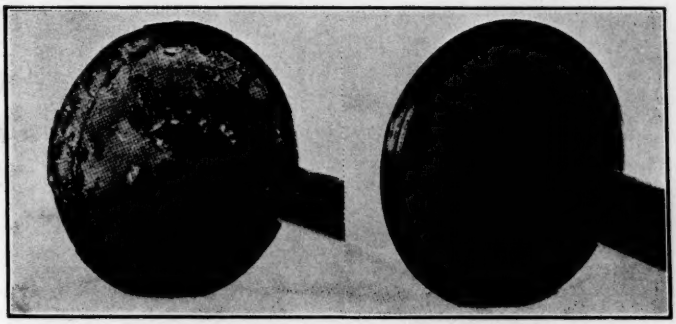
Even this, however, will be dispensed with or greatly reduced if all the claims and tests of a nickel chromium alloy recently placed on the market meets with successful confirmation in actual seagoing service in the oil engines of large and small motorships. We refer to Nichrome, a heat-resisting metal developed into valves under patents by the Driver-Harris Company of Harrison, N. J., and now available in large quantities after three years experimenting and exhaustive tests. We conscientiously advise all four-cycle engine builders to thoroughly test this alloy for valves.

So satisfactory and unusual has been the service given by Nichrome valves that they have been adopted as standard equipment by many companies. A test now under way on a large American Diesel vessel which has tried out other valves for which exceptional service has been claimed, shows Nichrome valves still in splendid condition, without any attention, after 15,000 miles, and with the engines running on boiler oils. The chief engineer of the motorship in question expressed enthusiasm to a representative of "Motorship" on the occasion of a recent visit. The engineers of the Winton Engine Works have also expressed their approval of Nichrome to us, having adopted these valves as a standard for all their Diesel engines.

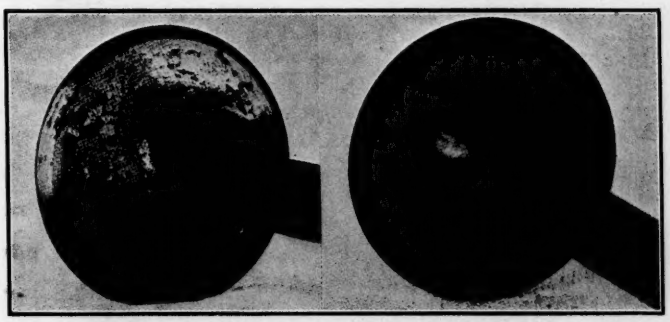
In view of the radical benefits that will be derived from the many adaptations of Nichrome if this new metal proves as satisfactory as tests and present commercial uses indicate, there is but little doubt that it will be widely adopted in this field. Meanwhile oil-engine builders will do well to thoroughly investigate its possibilities and make extended tests for themselves, both in the shops and in ships fitted with their engines in cylinders side by side with cylinders having other valves. We understand that the makers will be glad to make a Nichrome valve to the builder's drawings and furnish one for the purpose of a test.

Doubtless a few details regarding the history and development of Nichrome will be of interest. This basic alloy is not new. For more than fifteen years Nichrome, manufactured by the Driver-Harris Co., has been used as heat-resistors in electric-heating equipment. The present development and broad utility of electric heat, in both industrial and domestic applications, would have been impossible without an alloy that is highly capable of resisting heat and oxidation. Indeed, the electric heating industry could not have succeeded in any measure without such an alloy.

Its first mechanical adaptation and application in a large way was as heat-treating or carbonizing containers for case-hardening processes. In this field, as in the valve field, cast-iron had been used before this heat-resisting alloy was made available in the form of cast containers. The containers are subjected to high heat (1,500 degrees to 1,800 degrees F.) and to severe oxidizing conditions. The Nichrome alloy developed for carbonizing containers has not only overcome these conditions, but is totally



RESULT OF SIMILAR TESTS OF STEEL AND NICHROME (ON RIGHT) VALVES.



RESULTS OF SIMILAR TESTS OF TWO VALVES, ONE OF COBALT-CHROME AND THE OTHER (ON RIGHT) OF NICHROME.

immune to carbonizing, and is giving thousands of hours of uniform service. Cast-iron and other containers have been replaced in hundreds of plants throughout the country.

Being pioneers in the development and manufacture of heat and chemical resisting alloys it is altogether logical that attention should be turned by the makers to the poppet-valve question. This constitutes another development of the alloy Nichrome.

Engineers familiar with the functions of valves in internal-combustion oil-engines will immediately recognize the value of the characteristics outlined in the following, and the great advantages which valves possessing those characteristics, should add to even the very best designed engines. The points are therefore briefly stated:

Nichrome alloy for valves has a tensile strength of 20,000 pounds to the square inch at 900 degrees C.

At temperatures approximating 2000 degrees F. there is no scaling.

Highly resistent to chemical action of combusting fuel.

Will not absorb carbon.

The action of the valves does not injure valve-seats, because they are not extremely hard; but they are hard enough to withstand the shock of constant pounding.

Are made either in one piece or in two pieces with nickel-steel stem.

Hold compression to an unusual degree, because the heat of exhaust gases does not cause pitting to any appreciable extent.

Cannot be hardened or crystalized by heat treatment; retain their same physical properties as when installed. Therefore, they do not air harden.

The established facts which are cited above, are based on the performance of a considerable quantity of Nichrome valves made by Driver-Harris for use in Diesel-marine engines, for aircraft motors and automobile engines. So satisfactory have the results been over a long period of time, and even under adverse fuel conditions, that they have felt justified in providing additional manufacturing facilities so as to place Nichrome valves within reach of every owner and builder of internal-combustion engines.—"Motorship," Sept., 1922.

NEW DOUBLE-ACTING TWO-CYCLE DIESEL ENGINE.

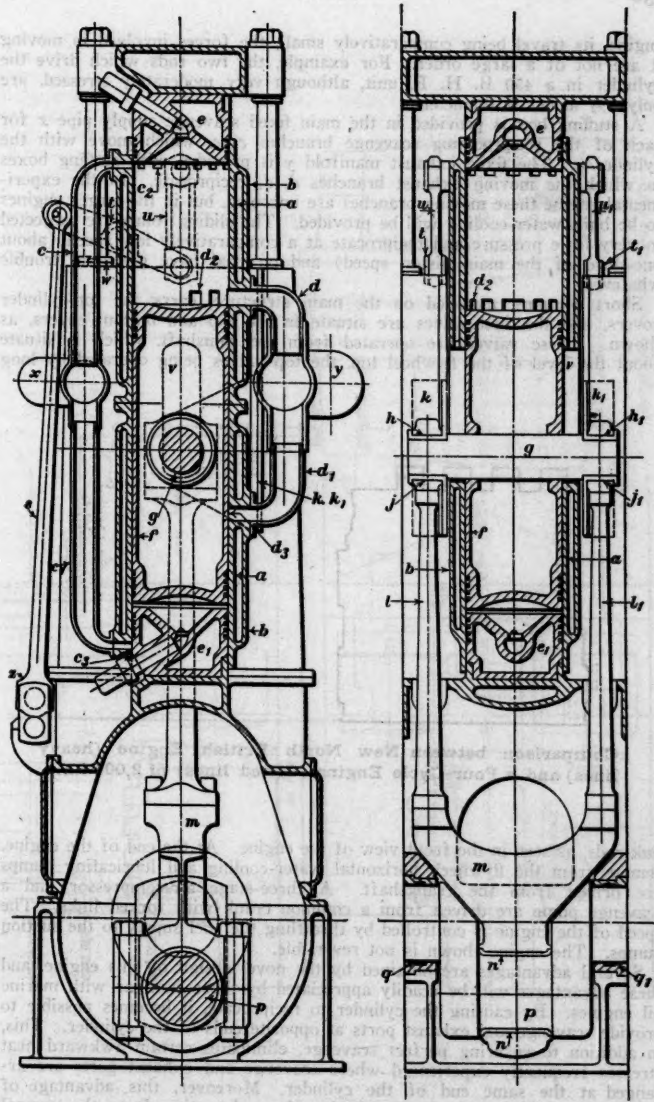
Interesting Tests at the North British Diesel Engine Works.

The marine Diesel engine as we know it today, whether it be operated on the four-stroke or the two-stroke cycle, suffers from certain inherent weaknesses, and although modern science and research have effected very substantial improvements during the last decade it cannot be said that finality has yet been reached. Not the least of these weaknesses lies in the fact that until recently, with one exception all marine Diesel engines were confined to the single-acting type. Early experiments with double-acting Diesel engines were not attended by a large measure of success, and the difficulties which beset the progress towards a satisfactory double-acting marine engine of the Diesel type have been abundantly manifest in the experience gained with the German-built vessel *Fritz*. The advantages which the double-acting engine possesses over the single-acting engine are many. Thus, a uniform turning effort is applied to the crankshaft and, approximately, double the power is developed in a given cylinder. Lubricating difficulties are greatly reduced by the reversal of pressure, the lack of which in some of the earlier Diesel ships which we have in mind was such an important contributory factor towards their failure. The double-acting engine, other factors remaining unaltered, should manifestly possess substantial ad-

vantages over the single-acting engine in the matter of weight and space occupied for any specific power, and given a sound design there is no reason why such should be more complex than that of the single-acting engine. Duplicate fuel valves at the bottom of each cylinder and the employment of scavenging valves in the case of the *Fritz* presented at the outset factors which mitigated against successful operation.

An entirely new type of two-stroke double-acting engine has been designed and constructed by the North British Diesel Engine Works, of Glasgow, and extensive tests have been carried out at Glasgow. These tests have given the builders such satisfaction that they are now engaged on the construction of a double-acting engine of 2250 I. H. P., to which we shall refer later. The most outstanding feature of this experimental engine has reference to the cylinder, which reciprocates with the engine, traveling through a distance approximately equal to one-third of the engine stroke. We publish herewith two views of this experimental engine, and we give diagrammatic outline drawings by the aid of which the principle on which the engine functions may be readily understood. In passing, it should be said that this engine has two working cylinders 11½ inches diameter by 14½ inches stroke and develops 240 B. H. P. at about 250 R. P. M. The engine is direct-coupled to an electric generator, and this engine and generator will shortly be installed as an auxiliary set in one of the large Diesel-engined ships at present being equipped by the builders. The engine consists primarily of an open-ended cylinder *a* which may be described as two cylinders bolted together. This cylinder, together with its water jacket *b* and scavenge and exhaust branches *c*, c_1, *d*, d_1 is free to reciprocate relatively to two cylinder covers *e*, e_1, provided at each end of the cylinder *a*. These covers are virtually stationary pistons and are provided with gas-tight rings so that the cylinder is permitted to move axially in relation to them. A piston *f*, on both sides of which the gases of combustion act, reciprocates in the cylinder *a* between the end covers *e*, e_1. Each cylinder cover is provided with a fuel valve and air starting valve. The piston *f* is provided with a gudgeon pin *g* which projects through axial slots *v* provided in the cylinder *a*. Crosshead guides *k*, k_1 are fitted to the ends of the gudgeon pin *g*. Bearings *h*, h_1 are provided on the parts of the crosshead guides which are attached to the end of the gudgeon pin *g*, and these bearings take the connecting rod-top end bushes *j*, j_1.

Connecting rods *l*, l_1 are provided between the crosshead bearings and the fork end *m*. A bottom end bearing *n* is provided at the lower end of fork end *m*, and this bearing operates on the crank *p*. Crankshaft bearings *q*, q_1 of the usual form are provided on either side of the crank *p*. The cylinder *a* and its water jacket *b*, together with the scavenge and exhaust pipes *c* and *d*, are caused to reciprocate by the crank motion *s*, *t*, *u*, *z*, the crank end of the auxiliary connecting rod *s* being attached to a crank *z* provided on the camshaft. The upper end of the connecting rod *s* is attached to the beam levers *t*, t_1 and the center gudgeon of the levers *t* is pivoted in brackets *w*, attached to the engine structure. Links *u*, u_1 are attached at their top end to trunnions provided on the cylinder, and at their lower ends to the levers *t*, t_1. The cylinder *a*, with its water jacket *b* and its scavenge and exhaust pipes *c* and *d*, travels generally in the same direction as the piston, but at a lower speed than that of the piston. Thus the scavenge ports c_2, c_3 which are supplied with scavenge air pipes *c*, c_1, are uncovered by the relative movement of the cylinder *a* to the cylinder covers *e*, e_1 and the exhaust ports d_2, d_3, through which the exhaust gases pass to pipes *d*, d_1, are uncovered by the relative movement of the piston *f* to the cylinder *a*. It should be noted that although the cylinder is a fairly heavy part of the

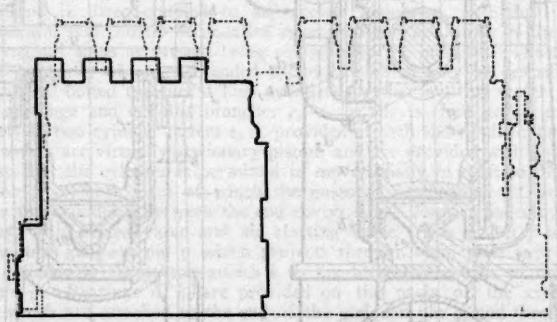


Sectional View of the New Double-Acting Two-Cycle Engine constructed by the North British Diesel Engine Works.

engine, its travel being comparatively small, the forces involved in moving it are not of a large order. For example, the two rods which drive the cylinder in a 450 B. H. P. unit, although very moderately stressed, are only 1½ inches in diameter.

A stuffing box is provided in the main fixed scavenge supply pipe x for each of the reciprocating scavenge branches c, c_1 which move with the cylinder a. The fixed exhaust manifold y is provided with stuffing boxes in which the moving exhaust branches d, d_1 reciprocate. In the experimental engine these moving branches are uncooled, but in the larger engines to be built water-cooling will be provided. The sliding joints are subjected to very little pressure and reciprocate at a comparatively low speed (about one-third of the main piston speed) and on trial have given no trouble whatever.

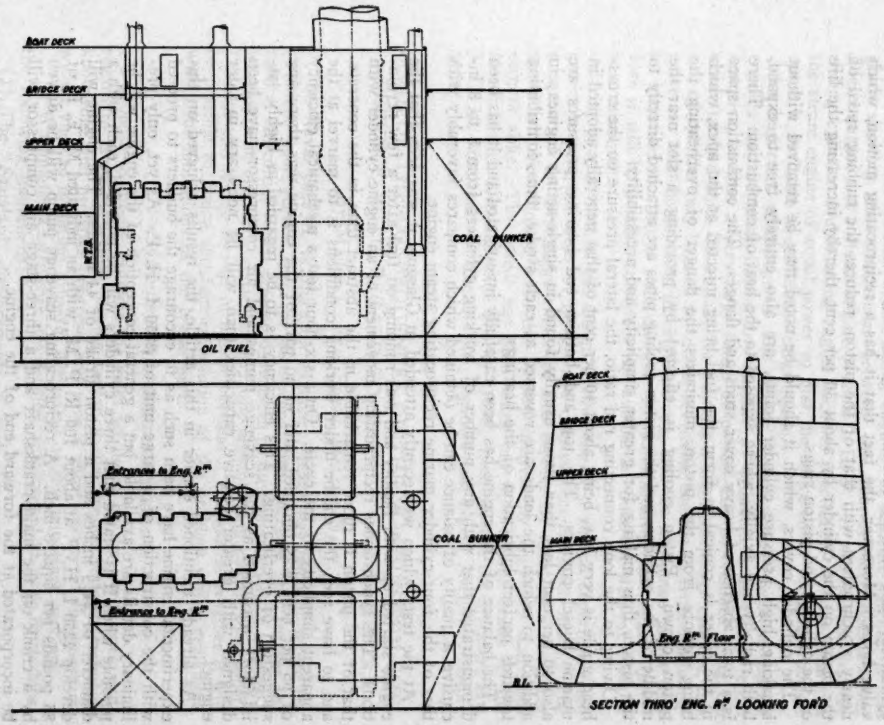
Short columns, mounted on the main structure, carry the top-cylinder covers, and the fuel valves are situate in the top and bottom covers, as shown. These valves are operated from the camshaft, which is situate about the level of the flywheel top, the top valves being operated by long



Comparison between New North British Engine (heavy lines) and a Four-Cycle Engine (dotted lines) of 2,000 i.h.p.

link-rods, as seen in the front view of the engine. At the end of the engine, remote from the flywheel, horizontal water-cooling and lubricating pumps are driven from the crankshaft. A three-stage air-compressor and a scavenge pump are driven from a common crank with forked links. The speed of the engine is controlled by throttling the fuel supply to the suction pumps. The engine shown is not reversible.

Several advantages are obtained by the novel design of this engine, and these advantages will be readily appreciated by those familiar with marine oil engines. By causing the cylinder to reciprocate, it becomes possible to provide scavenge and exhaust ports at opposite ends of the cylinder. This, in addition to ensuring perfect scavenge, eliminates certain awkward heat stresses frequently experienced where scavenge and exhaust ports are arranged at the same end of the cylinder. Moreover, this advantage of "through" scavenge has been gained without departing from the normal well-tried form of crankshaft.



Comparison between a North British Double-Acting Engine (heavy lines) and a Steam Set (dotted lines) of 2,000 i.h.p

The engine is particularly well suited to accommodate expansion due to heat; one of the most serious problems with which Diesel engine designers are faced today. The cylinder is entirely free to expand, both axially and radially. In way of the combustion space, it takes the form of a plain cylindrical tube entirely surrounded by cooling water and free from any valve holes. Moreover, the fact that it has a reciprocating motion, which nearly synchronizes with that of the piston, reduces the rubbing speed of the piston on the cylinder by about 30 per cent, thereby increasing the life of the cylinder and piston rings.

The cylinder covers, which it should be noted may be removed without breaking high pressure cylinder joints, are also entirely free to expand, both radially and axially, where exposed to the heat of combustion. There is a total absence of heavy cover studs and flanges. The combustion space in the cover is conical in form, the fuel being injected at the apex, which being remote from the piston, minimizes the danger of overheating the piston crown. Piston cooling is effected by providing a slot near the middle of the cylinder, and the piston cooling pipes are attached directly to the piston, thus making for greater simplicity and accessibility.

Owing to the long connecting rod ratio, the lateral pressure on the cross-head guide is very low, being about 40 per cent of that generally adopted in marine Diesel engines. The top and bottom end bearing pressures are also 40 per cent less than those usually found in single-acting engines; in addition to which the loads are reversed at each stroke, thus contributing towards perfect lubrication of the bearings.

The balance of the engine has been carefully investigated and it has been demonstrated that with any number of working cylinders from 3 to 8 inclusive, a quality of balance can be obtained which compares favorably with that of the four-cylinder marine reciprocating steam engine.

At the tests which we recently attended in Glasgow it was seen that the engine started up with facility and was running on fuel after a few revolutions. The synchronous reciprocating movement of an engine cylinder with that of the piston is more remarkable in the abstract than in the concrete, and to have seen the engine under working conditions is to marvel at the apparent simplicity of the cycle. On a six-hour test a mechanical efficiency of 65 per cent was obtained, and in no part of the engine was there any suggestion of overheating. This efficiency is to be regarded as highly satisfactory inasmuch as the scavenge pumps and air compressor have been designed with greater relative capacities than will be necessary in later engines.

As already mentioned earlier in this article, the results achieved on this experimental engine have been such as to encourage the builders to proceed with the construction of a large unit of 2250 I. H. P. As yet, only preliminary details are available, but a general conception of the engine will be possible when it is stated that three cylinders will be provided, each with a diameter of 24½ inches and a piston stroke of 44 inches. The engine will develop 2250 I. H. P. at about 100 R. P. M., with an indicated M. E. P. of 86 pounds per square inch. A reciprocating scavenge pump will be driven by a crank on the main crankshaft and a three-stage air compressor will be incorporated at the forward end of the engine.

Diagrams are given illustrating comparisons between a four-cylinder double-acting engine of this type, developing 2000 I. H. P., and the four-cycle single-acting Diesel engines as installed in the *Domala*. The latter engines were also manufactured by the North British Diesel Engine Works.

The four-cylinder double-acting engine has a 22-inch bore cylinder against 26½ inches in the four-cycle, and weighs exactly 50 per cent less than the

four-cycle engine. It has 50 per cent fewer main bearings, bottom end bearings, crosshead bearings, and guide shoes, and has only eight cylinder valves in operation at sea against 24 in the case of the four-cycle engine. Moreover, the four-cycle engine has 88 per cent greater piston rubbing speed than the sliding cylinder double-acting engine. A separate diagram is also given showing the comparison between the four-cylinder double-acting Diesel engine and a steam engine of 2000 I. H. P. It will be seen that the sliding cylinder double-acting engine occupies practically the same space as the steam engine of equal power so that the entire boiler room and bunker space is available as additional cargo space.

An important advantage of this engine is that direct reversible marine engines may be made having any number of cylinders from two to eight inclusive, so that a very large range of powers can be obtained using the same standard unit. If we take for example a 450 horsepower cylinder, this size of cylinder may be used for direct-reversible engines of from 900 horsepower to 3600 horsepower in single screws, and the maximum power may be increased to 7200 in twin screws. In manufacturing this engine therefore it will be possible to reduce the number of different sizes of cylinder to a minimum.

It is apparent that in the design of this engine a radical departure from generally accepted ideas of design and construction has been made. The progress of the first large engine of this type to be fitted in a sea-going vessel will be followed with the closest interest by engineers and shipowners alike. The patent has been taken out in the joint names of the North British Diesel Engine Works (1922) Limited, and the inventor, Mr. J. C. M. Maclagan.—"The Marine Engineer and Naval Architect," Sept., 1922.

SOME OF THE CONSEQUENCES OF THE WASHINGTON CONFERENCE WITH REGARD TO NAVAL CONSTRUCTION.*

By Major L. Fea, Royal Italian Navy.

The Naval pact of Washington (January, 1922) has imposed severe restrictions upon certain types of battleships; they increase still further the natural difficulties which present themselves in the construction of a modern warship unit, and, as a matter of fact, the present requirements of a super-Dreadnought are so serious in themselves that it is doubtful whether it will be possible to find a satisfactory solution of the problem whilst remaining within the limitations of the Naval pact.

WASHINGTON CONFERENCE DECISIONS.

I propose to briefly examine the rules set down by the Conference with regard to large battleships, having regard to the present state of technical knowledge and probable progress, and give a short survey of the question, with a view to initiating a useful discussion of this important problem.

The Washington Conference has decided, with regard to large battleships, the following limitations:—

(1) The "Standard" displacement† of each ship must not exceed 35,000 tons.

(2) The maximum caliber of the guns must not exceed 16 inches.

(3) Ships shall not be replaced under 20 years.

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* Abstract of Paper read at the Paris Summer Meetings of the Sixty-third Session of the Institution of Naval Architects, July 4.

† The Conference "standard" displacement does not include fuel and feed water.

(4) The "Standard" total tonnage of battleships of the various Powers shall be as follows: Great Britain and America, 525,000 tons each; Japan, 315,000 tons; France and Italy, 175,000 tons each.

Incidentally it may be said that the last clause will greatly increase the difficulties of the smaller navies, for it is more difficult to follow up progress and construct a homogeneous fleet, when it is only possible to build a ship every four years, than if one is allowed to build two or three at the same time.

The details of battleships built and in course of construction after the Battle of Jutland, tend to show that the "Standard" displacement of 35,000 tons has already been largely surpassed; the *Hood*, the *Massachusetts*, the *Tosa*, all exceed 40,000 tons displacement, although the American and Japanese ships have the moderate speed of 23 knots. This proves that today it is impossible to unite all the requirements of a modern ship under 40,000 tons; on the contrary, as has been indicated by Sir G. Thurston, one requires to have 55,000 tons, if the locks of the Panama Canal were sufficiently wide.

The limit of 55,000 tons would not seem exaggerated if we consider that two 16-inch 45-caliber guns with their turret weigh more than 1500 tons, that is to say, as much as the total armament of a battleship (pre-Dreadnought)—that the average weight of propelling machinery of 50,000 horsepower is more than 2000 tons—that the armor alone of a modern ship weighs as much as the "Standard" displacement of a pre-Dreadnought.

This shows that for this reason the 35,000 tons of the Conference is not sufficient, if the problem of the present-day battleship is to be solved "without compromises." Is it possible to make better provision in view of possible progress of technical knowledge, or on the basis of acceptable modifications in the actual requirements of modern battleships?

OFFENSIVE POWER.

After the Washington Conference it evidently will not be possible to adopt guns of less than 16-inch caliber.

Whatever may be the definite results of investigating the effect of long-range shots with heavy projectiles and of attacking ships with poison gas, it is not impossible, especially in Southern seas, that new means of fire-control and spotting may enable the range to be still further increased, considerably beyond the horizon.

If we assume that the principal caliber is 16-inch, we must choose the best number of guns to be put on board, keeping before us the necessity of single salvos and also of successive salvos.

Calculation shows that the probability of hitting grows with the number of shots in each salvo, first rapidly and then more slowly; one can be more or less sure that with between six and nine shots per salvo one can reach the maximum probability of hitting. Therefore it is sufficient if each salvo has six to nine shots.

The interval between each salvo is related to the rolling period of the ship (at least until we find some new mechanical means of firing), a period which varies in modern ships from 12 to 14 seconds (complete double oscillation).

It is therefore evident that the number of guns on board a ship depends, the number of shots per salvo remaining constant, on the quickness of loading. If it were possible to reduce the speed of loading of all guns below the period of complete rolling, or better, below the half period of rolling, it would be possible to reduce the number of guns on each ship to correspond

with the minimum number of shots necessary for each volley. This would reduce the number of large caliber guns on board our ships by about half, without in any way interfering with their offensive power.

SECONDARY ARMAMENT.

Although it appears possible, though not in fact already achieved, to arrive at such a reduction in the primary armament, we do not reach the same conclusion with regard to the secondary armament, destined for fighting smaller warships and aircraft. It is possible to have a single type of mounting, naval and anti-aircraft, in the guns of medium caliber up to 6 inches, thus uniting the anti-aircraft and anti-submarine guns. Therefore the actual number of guns in these batteries should be increased from the present 12 to 16, and even more, so as to have for each quadrant a number of pieces not less than four. It is also necessary to preserve these minor armaments in good condition during and after the battle; the practice of not protecting these guns, as in England and America, would not seem advisable. It appears indispensable to fit them with turrets designed in such a way as to ensure rapidity of fire and the necessary elevation, or to adopt the system of well armored disappearing mountings.

Naturally this protection must be proportioned to the attack and not to the caliber of the guns thus protected. Therefore, rather than consider a reduction in the weight of secondary armament, we must frankly face a considerable increase in this weight.

In conclusion, as regards offensive power, we may say that the only direction in which we can hope to effect the reduced displacement of future battleships is by a radical modification of the heavy artillery, which would allow of a diminution, to a certain extent, in their number. Réduction in the lesser armament and in under-water protection does not seem either possible or advisable.

SPEED.

What is understood today as "high speed" is quite easy to comprehend; a speed of 30 knots is already exceeded, and it is not possible to go back. If obvious conclusions, such as the consequences of very high speed on various kinds of ships have any weight, it might be advisable for a large navy to keep below the above limit, as, it would appear, the British Admiralty are doing after the example of the *Hood*. Can a navy with a limited number of ships renounce the only strategical and tactical advantages, which, with the same gun power, can be drawn from possessing a higher speed? This is certainly the most difficult and pre-occupying point in the problem of future warships, as upon the more or less satisfactory solution of this problem the success of an entire navy may depend for years to come.

We may look ahead and observe the results which might be obtained with internal combustion motors, with heavy and reliable fuel. The other element, allied to speed, is autonomy; the Washington Conference by fixing, as already stated, the "Standard" displacement of battleships has taken away considerable importance from this element, naturally to the advantage of the ships destined to fight on the ocean.

PROTECTION.

Let us see if the modern science can give us that defense which will be necessary against attacks in three directions: side protection, deck protection, and under-water protection.

For the side protection we must still have recourse to massive external defenses. The experiments carried out by the British Admiralty with the ex-German battleships have shown that everywhere technical science has reached the same state of perfection; marked progress in this direction does not seem probable in the near future, unless unforeseen circumstances arise.

As a defense against 16-inch shells, plates of equal thickness would probably be necessary, even at long ranges. This fact, in conjunction with the limitations of Washington, imposes on everybody the necessity to return to the old principle, to which the Americans were the first to revert, that is to say, to limit the vertical protection to the essential parts, reducing it to armor of medium thickness, which would be of no use against attacks of heavy artillery. Nevertheless, the armor, naturally, must always be *sufficient*. Therefore, even by reducing the vertical protection to the indispensable minimum, no great advantage as regards weight can be obtained.

The utmost importance is given, and rightly, to the deck protection, as the horizontal target is by far the greater, not only in aerial attacks, but also against artillery at the actual ranges of battle.

Whilst we may say as a sure guide that it is indispensable to considerably increase the thickness of the protection to decks, it would seem that some doubt still exists as regards the allocation of thickness in this protection.

It would be advisable to sacrifice the part of the ship above the principal protective deck as long as one can, at any cost, save the one underneath. For this reason horizontal protection will require greater sacrifices, and will not allow any margin of economy.

UNDER-WATER PROTECTION.

The under-water defense is now of fundamental importance, and is allied to the strength and stability of the ship. The three known fundamental solutions of the problem are: the bulge system adjoining the keel, proposed and experimented upon by General Ferrati in 1911, and largely used in the British Navy; the system of watertight compartments, investigated in 1892 by the late General Benedetto Brin, which is largely used in the American Navy; and the system of "hydrodynamic action" of Colonel Pugliese.*

These examples of under-water protection are sufficient to enable one to understand that in this respect also it is not possible to anticipate notable reductions in the matter of weight.

A further system of defense will in future have to be provided, that is, against poisonous gas, whatever may be the consequence of the Washington Conference as to the abolition of this gas in future wars.

This is an entirely new question, which will have to be thought out from the beginning of the building of new ships, so that the security obtained against this kind of attack may not be inferior to that provided against other kinds. It is difficult to estimate exactly what increase in weight this protection against poisonous gas may involve, but it is certain that it will not be small.

All forms of protection are now assuming great importance, as shown by the fact that on a ship like the *Hood*, which is essentially an attacking ship, the weight of the protection alone is practically three times as great as the weight devoted to the offensive.

It is difficult to foresee whether, in view of the offensive and defensive conceptions, which we have here set out, it will be possible to balance the one with the other so that they shall not cause an increase in the total dis-

* See *Rivista Marittima*, Vol. LIV.., No. 7, page 67.

placement. But this result, which to us would seem satisfactory, and which probably might be carried out, would not suffice to come within the limitations of the Washington Conference.

POSSIBLE CONSTRUCTIONAL IMPROVEMENTS.

As regards the intrinsic qualities of the ship, advantages can only be obtained by means of steady and continual perfectioning of the design and material. These improvements might, amongst others, relate to—

1. The use of more perfect materials.
2. The general use, if possible, of welding instead of riveting.
3. The incorporation of the protective elements with the structural ones.

As regards the first point, it would seem possible to obtain notable results, apart from the question of expense, and always with a view to limiting the displacement, by the use of the special steels, better adapted than those actually in use, to resist tension, and above all the strain (alternating and dynamic strains), Vanadium and Molybdenum already used for armored plates, which is a magnificent material for construction. The result of adopting such a material would be remarkable, as almost 60 per cent of the hull weight in a modern ship is devoted to longitudinal or transverse resistance.

As regards the second point, both the difficulties and the advantages of welding, as against riveting, are well known. Today the welding of mild steel can be regarded as perfectly reliable, but as regards special steels, especially those above mentioned, grave doubts probably exist. In resistance of the structure against explosion, under-water or by aerial bombs, the welding process has an influence of first-class importance.

PROTECTION AND STRUCTURAL STRENGTH.

The third point, however, is, in our view, the one which demands most attention, as it is the one which contains potentially the greater promise for the future. The co-operation of the protective material with that providing structural strength is already partly achieved in several navies, so much so that in certain of them the conventional weight of the *hull* includes part of the protective elements of the armored decks.

This principle has had an important application in the recent German light cruisers, in which the sides were constructed of armor plate of a thickness of about 60 mm., embodied very cleverly with the rest of the structure, and with the various plates. The protective under-water structures, composed of one or more watertight compartments, flat or curved, and of great thickness, were an integral part of the hull, and they contributed greatly to the general structural strength.

The advantage which may be gained by an extensive use of this system is evident, as we know how large a part of the total weight the following items constitute: hull plus armor (in the *Hood*) amount to 71.7 per cent, while in the Italian pre-Dreadnought ship *Regina Margherita* it is only a little different, viz., about 70 per cent. Even a slight economy in this direction would give a considerable saving. If one were to obtain, for example, a reduction of 10 per cent for a large modern ship, where hull and armor amount to about 20,000 tons, one would save a weight equal to that of an entire triple turret of 16-inch guns.

In any case, three points of such importance to naval engineering, to which we have referred, would produce results which can only be attained slowly, and as a consequence of investigation and experiments which will require considerable time.

CONCLUSION.

The analysis which we have made regarding the various qualities, naval and technical, of the modern battleship does not leave us any illusions as regards the impossibility of being able to comprise within the limits of displacement fixed by Washington all the offensive and defensive power which at the present day would seem desirable. It would be well to note that, whatever may be the road which it will be necessary to follow in order to gain this object, two conditions must be observed: a systematic and profound study of all the material which is necessary for the construction of such a complex machine as the modern warship; and very great expense, which may cause one to doubt whether the ship of the future of 35,000 tons imposed by the Washington Conference will really cost the various Governments much less than the present-day ship of 50,000 tons.—"Shipbuilding and Shipping Record," July 6, 1922.

FIRST MERCURY BOILER AND TURBINE BEING MADE HERE.

The first commercial mercury boiler and turbine in the world will shortly be shipped from Buildings 52, 60, and 105, and will be installed at the Dutch Point generating station of the Hartford (Conn.) Electric Light Company. The mercury turbine unit is about 2000-kv-a. capacity and every facility will be provided for analyzing its performance. Fuel oil will be burned and will be weighed instead of measured; recording instruments will be installed to enable the energy consumption of auxiliaries to be studied, and feed water and other items will be measured. The equipment will be housed in a 26-foot by 45-foot structure now being erected by Stone & Webster outside the plant proper. It is hoped to cut down the fuel consumption per kilowatt-hour by approximately 50 per cent through the use of the mercury turbine and utilization in steam production of heat given up in the mercury condensation process.

The general arrangement of the apparatus is to install a 2000-kv-a. mercury turbo-generator on the top of an oil-fired mercury boiler containing a charge of about 30,000 pounds of mercury. The mercury vapor at rather low pressure operates the turbine. The exhaust from the mercury turbine produces steam in a modified water boiler, and this steam will be introduced into the steam lines of the plant.

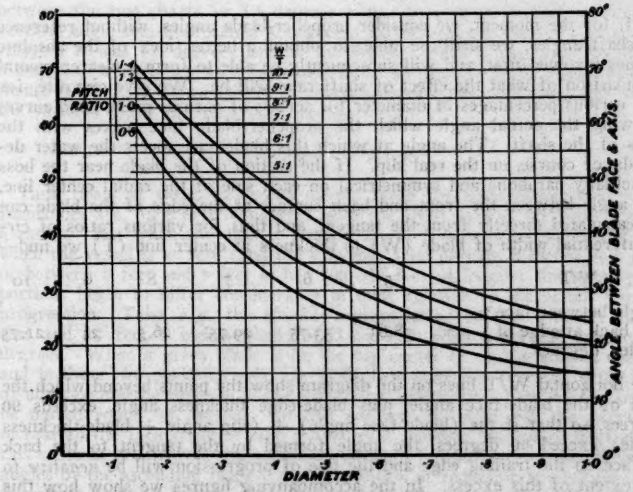
W. L. R. Emmet, Consulting Engineer of the General Electric Company, has devoted years to the development of the mercury boiler and turbine and this first installation will be keenly watched by him. Mr. Emmet has been with the General Electric Company since its formation in 1892, and has been closely identified with the development of the Curtis turbine industry. Among the many inventions standing to his credit are the oil switch, varnished cambric cable, several types of transformers and insulation for alternators. A great deal of Mr. Emmet's most useful work cannot, however, be classified.

Years ago Mr. Emmet conceived the idea of electric propulsion for ships and began a vigorous campaign for its adoption. A long period intervened before Mr. Emmet's perseverance culminated first in the collier *Jupiter*, and later the U. S. S. *New Mexico*.—"Schenectady Works News," July, 1922.

THE RAKE OF PROPELLER SHAFTS.

FACTORS WHICH INFLUENCE SHAFT RAKE.—EFFECT OF RAKE ON PROPELLER EFFICIENCY—VERTICAL AND HORIZONTAL ANGLES.

The principle of slightly raking propeller shafts in the vertical or horizontal planes besides having much to recommend it, is often inevitable and may be said to be either advisable or necessary for the following reasons: the former in order to suit the intended form of ship, which really means the choice of suitable propeller position at one end of the shaft and the convenience of machinery installation at the other, while as regards the latter, the necessary clearance between hull and propeller blade tips, plus the half diameter of the propeller to the shaft axis, may enforce an outboard bearing center which leaves the designer no option but to install a shaft angle greater than that which would otherwise be acceptable.



Curves showing Angle between Propeller Blade Face and Shaft Axis.

EARLIER PRACTICE.

There is no doubt that in many recent cases of high-powered twin-screw torpedo boat destroyers the shape of stern of ship, the propeller diameter necessary to absorb 22,000 S. H. P. per screw, and the fact that the use of single-reduction gear wheels involves a high shaft center in the engine room have all combined to enforce very steep shaft angles. In single-screw vessels, wherein the after-shaft center is settled by the propeller diameter, and the distance of its tip above the keel line, the only object in departing from the horizontal is to reduce the height of shaft tunnel in the case of reciprocating engines, or to admit of automatic cylinder drainage in the

case of turbines, but in the case of twin, triple and quadruple-screw vessels, both vertical and lateral divergence may become a matter of considerable importance. For instance, with triple shafts and direct-connected turbines it was usual to adopt center lines parallel with that of the ship, and to allow only such vertical inclination as would clear the turbines of water, whereas with triple shafts and reciprocating engines a very considerable divergence of outer shafts has generally been necessary in order to avoid the overlapping of the several propeller discs. With twin-screws and reciprocating engines little or no difficulty has usually been experienced in selecting suitable positions of screw from the point of view of hull efficiency, tip clearance and flow of water past bossing or brackets, and of machinery without the use of angles exceeding one or two degrees, but the increase in power per shaft which we have recently witnessed has had the effect of increasing these amounts to a marked extent, particularly as regards the vertical angle.

PROPELLER-BLADE ANGLES.

If, for the moment, we consider propeller-blade angles, without reference to shaft angles, we shall be able to obtain a better idea of the absolute values involved first, and will subsequently be able to form a clearer mental comparison of what the effect of shaft rake will be. We give opposite, for the various percentages of diameter for screws of various pitch, ratio curves showing the actual angle which the propeller-blade face makes with the axis of the shaft. The angle at which this blade face meets the water depends, of course, on the real slip. If the section of the blade near the boss is actually parabolic and symmetrical on each side of the radial center line, the angle between the front and back surface at the edge of the blade can be calculated directly from the tangent, and thus for various ratios of circumferential width of blade (W) to thickness at center line (T) we find—

W/T	=	5	6	7	8	9	10
Angle between face and back at edge of blades, degrees	=	38.63	33.75	29.75	26.5	24	21.75

The horizontal W/T lines on the diagram show the points beyond which the sum of the blade-face angle, plus blade-edge thickness angle, exceeds 90 degrees, so that if the (blade-face angle) + (slip angle + blade-thickness angle) exceed 90 degrees, the angle formed by the tangent to the back surface on the trailing edge and the line of progression will be *negative* to the extent of this excess. In the accompanying figures we show how this occurs by representing a thin modern destroyer propeller and the propeller of the late Hamburg-America Atlantic liner *Deutschland*. Both sections are shown at one-third of the screw's radius, and the center line is taken as parallel with that of the ship. Now, the horizontal divergence is generally very small, and only in a few extreme cases of twin-screw turbine-driven torpedo craft in which separate engine rooms have been placed fore and aft do we find the angle between shaft and ship as much as 1.5 degrees.

TENDENCY OF VERTICAL RAKE.

Vertical rake, however, has been greatly increased in recent years. When the direct-connected turbine was first installed in torpedo craft, very high revolutions and small diameter screws and small overall diameters of turbine cylinders were the rule, and little inclination was needed. Thus, in the *Tribal* class, the angle between shaft and water line was about 2.7 degrees on

the center and 3.7 degrees on the wing shafts, and in the succeeding class—the *Beagles*, also with triple screws—it was only about 2.0 degrees. In the later twin-screw designs, when the propeller and turbine diameters were increased roughly in the ratio of $\frac{\sqrt{3}}{\sqrt{3}}$, the engine-room center rose and the screw axis had to drop, with the result that angles of 3 degrees-3.5 degrees rapidly increased to 4 degrees, 4.5 degrees and even to 5 degrees when the main turbines were placed side by side, but when, as in the German torpedo craft, where the standard engine-room arrangement consisted of the starboard turbine in one room forward, and the port turbine in a separate one aft, the short shaft-length of the latter forced up the angle of descent considerably, and in successive pre-war years the port and starboard angles became 4.3 degrees and 2.1 degrees, 5.4 degrees and 2.3 degrees, 5.7 degrees and 2.7 degrees, respectively, the horizontal divergence totalling 3 degrees between the two shafts, or 1.5 degrees each. In several British vessels the vertical angle had approached 4 degrees; in the case of the "V" class destroyers it was 4.6 degrees, and in the *Mackay* class it amounted to about 5 degrees, or as much as 1 inch per foot. In the U. S. S. *Pruitt*, whose thrust trials have frequently been compared with those of the latter ship, the vertical angle was only 2.56 degrees.

VARIATION OF PITCH.

Tank trials of series of model screws made for ascertaining basis performances are, of course, conducted with the shaft axis in the line of progression. But when the axis is already set at 5 degrees to the horizontal when the vessel is at rest (and the ship may trim perhaps 2 inches in 6 feet or between 8 feet and 9 feet in her 300 odd feet of length) the screw proportions begin to differ considerably in their relation to the actual line of progression. Take, *e.g.*, the *Mackay's* propeller of 10 feet 6 inches diameter and 12 feet 6 inches pitch; P/D = 1.19, with a vertical rake of 5 degrees. When a given blade is on the top center or on the bottom center, and in these two positions only, the blade-face angles in relation to the line of advance are the same as if the shaft were horizontal. Now let the screw be moved round 90 degrees. In this position, the rake has the effect of *increasing* the face angle at each radius by a constant amount—5 degrees in this case: at 270 degrees the same amount must be *subtracted* from the pitch angles of the ascending blade. As the addition or reduction is constant over the whole blade, it follows that, for the descending blade the pitch becomes, in relation to the actual direction of motion of the screw (for its relation to its axis obviously remains unchanged, except for any possible springing of the blades), an outwardly increasing pitch; for the ascending blade it becomes an outwardly decreasing amount. If for the *Mackay's* propeller at 90 per cent of the radius we refer to Fig. 1, we find a blade-face angle of 23 degrees with the axis. At 90 degrees and at 270 degrees the angle becomes 28 degrees and 18 degrees, respectively, and converted into terms of pitch (at 0.9 degree of the radius) we have—

Blade at top center with face pitch	=	12.6
Blade at 90 degrees center with face pitch	=	15.9
Blade at 180 degrees center with face pitch	=	12.6
Blade at 270 degrees center with face pitch	=	9.6

The normal full-speed revolutions are practically 360 per minute, or 6 per second, so that the variation from normal to maximum takes place in 1/24th second and from maximum to minimum in 1/12th of a second. The variation from the normal pitch ratio of 1.19 is from 0.905 to 1.5 at this radius: it varies, of course, with the radius as will be seen from Fig. 1.

EFFECT OF RAKE ON PERFORMANCE.

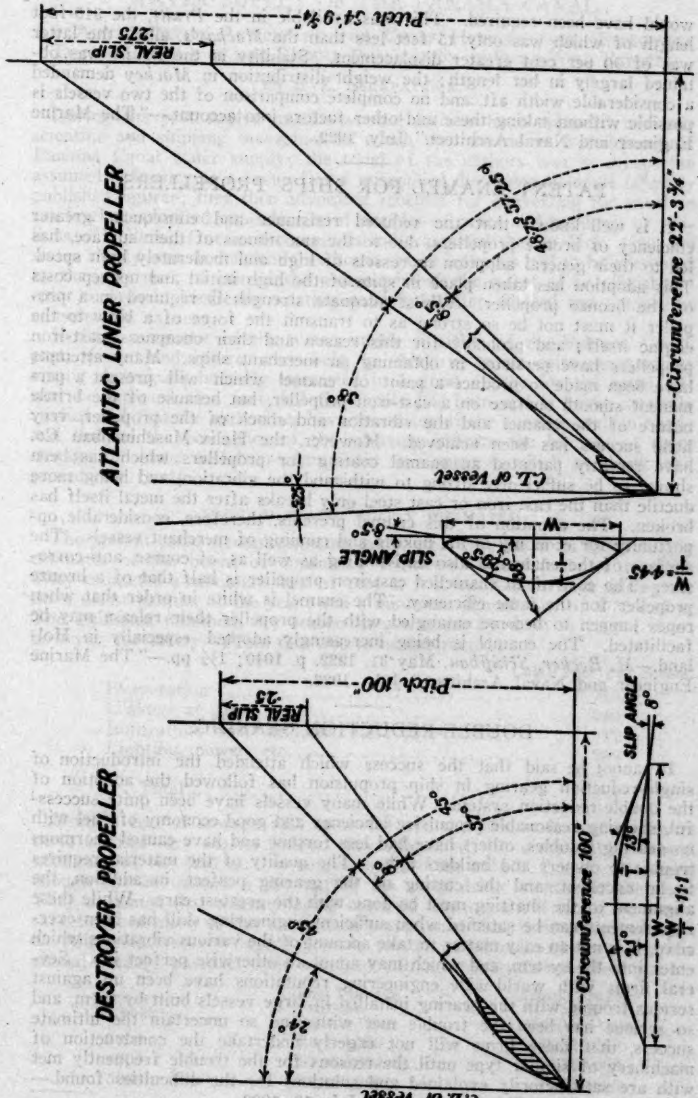
It is interesting to consider what this variation involves: it obviously upsets all calculations of pitch, slip, pressure, &c. The effect is to throw the center of thrust off the center of the shaft and the consequence is an increase of bearing pressure in the A bracket bearing. This is essentially a very thoroughly water-cooled bearing (forced lubrication has often been applied to these bearings as the result of experience), and there is no means of knowing how much power is wasted in friction at this point, but it is hard to believe that it is not a very tangible quantity. Taking the figures for *Mackay's* light-load trial, speed of ship = 35.46 knots, and R. P. M. = 353.4, the pitch limits indicated show a variation in real slip between 0.012 and 0.405.* If the variation of pitch be worked out for the U. S. S. *Pruitt* at the corresponding position on the screw, it will be found that, from the blade-face angle of about 21.3 degrees, the total variation is from 18.7 degrees to 23.9 degrees, compared with 18 degrees to 28 degrees from the 23 degrees of the *Mackay*.

However much independent analysts may differ in their own interpretation of the thrust trials of these two vessels as regards the balance between water factor and the corresponding analysis pitch factor taken, all will agree that in relative rotation efficiency the *Pruitt* was markedly superior to the *Mackay*. There can be little or no doubt that this extreme rake in the case of the latter must have exercised a very material influence on the performance of the screw for more reasons than one. A propeller, like many other things, has two faces, and the same addition or subtraction of angle of rake must be made to both sides. The variation of actual pitch (not face pitch) over the blade therefore becomes almost insolubly complex. But there is also one further objection to this extreme rake. The long and heavy A bracket boss presents a considerable projected surface to its line of progression through the water, and passing through it at full speed—some 60 feet per second—it must obviously render the mass of water into which the propeller enters far less dense than would be the case if the shaft axis and line of progression were more nearly in the same plane. In other words, it largely nullifies the suction action on the back of the blades; it adversely affects the quantity of water which should pass through the screw's disc, and further prevents the projection of a mass of water astern which is part of the proper function of a propeller in obtaining the necessary reaction to drive the ship.

EXTREME RAKE A DISADVANTAGE.

It is, however, impossible to dissociate this particular disadvantage with the design of the ship as a whole. In order to obtain a flow of water more at right angles to the disc of the propeller, such as was actually secured in the case of the *Pruitt*, a considerable modification to the run of the hull

* In this connection, the writer in an entirely independent analysis to that of Mr. Frodsham Holt, extracted a much lower (Froude) wake factor for this trial. The above figures follow Mr. Holt's own table using B = .12 (*i.e.*, not B × 0.925). With the lower wake factor (= 0.035) the real slip at the 270 degree position is actually about 0.03 ***negative***, and about 0.38 at the 90 degree position.



Comparison between Destroyer and Atlantic Liner Propeller Blade Angles.

would have been required. This was available in the *Pruitt*, the 310-foot length of which was only 15 feet less than the *Mackay's*, albeit the latter was of 50 per cent greater displacement. Stability in the *Pruitt* was obtained largely in her length; the weight distribution in *Mackey* demanded a considerable width aft, and no complete comparison of the two vessels is possible without taking these and other factors into account.—"The Marine Engineer and Naval Architect," July, 1922.

PATENT ENAMEL FOR SHIPS' PROPELLERS.

It is well known that the reduced resistance and consequent greater efficiency of bronze propellers, due to the smoothness of their surface, has led to their general adoption in vessels of high and moderately high speed. This adoption has taken place in spite of the high initial and upkeep costs of the bronze propeller. Whilst adequate strength is required in a propeller it must not be so strong as to transmit the force of a blow to the engine itself; and probably for this reason and their cheapness, cast-iron propellers have persisted in obtaining on merchant ships. Many attempts have been made to produce a paint or enamel which will present a permanent smooth surface on a cast-iron propeller, but because of the brittle nature of the enamel and the vibration and shock of the propeller, very little success has been achieved. However, the Helix-Maschinenbau Co. have recently patented an enamel coating for propellers which has been shown to be sufficiently strong to withstand the vibration, and being more ductile than the cast iron or cast steel only breaks after the metal itself has broken. The adoption of this enamel presents, therefore, considerable opportunity for economy in the upkeep and running of merchant vessels. The surface of the enamel is also anti-fouling as well as, of course, anti-corrosive. The cost of an enamelled cast-iron propeller is half that of a bronze propeller for the same efficiency. The enamel is white in order that when ropes happen to become entangled with the propeller their release may be facilitated. The enamel is being increasingly adopted, especially in Holland.—*M. Becker, Schiffbau*, May 31, 1922, p. 1040; 1½ pp.—"The Marine Engineer and Naval Architect," July, 1922.

DOUBLE-REDUCTION GEARING.

It cannot be said that the success which attended the introduction of single-reduction gearing in ship propulsion has followed the adoption of the double-reduction system. While many vessels have been quite successful, showing reasonable propulsive efficiency and good economy of fuel with no gearing troubles, others have had less fortune and have caused enormous trouble to owners and builders alike. The quality of the material requires to be excellent, and the cutting of the gearing perfect, in addition, the alignment of the shafting must be done with the greatest care. While these requirements can be satisfied when sufficient engineering skill has been exercised, it is not an easy matter to take account of the various vibrations which enter into the system, and which may annul an otherwise perfect job. Several firms with world-wide engineering reputations have been up against serious trouble with the gearing installed in large vessels built by them, and so serious has been the trouble met with, and so uncertain the ultimate success, that these firms will not eagerly undertake the construction of machinery of similar type until the reasons for the trouble frequently met with are satisfactorily explained and solutions for the difficulties found.—"Shipbuilding and Shipping Record," July 13, 1922.

WATER SUPPLY OF THE PANAMA CANAL.

AN AUTHORITATIVE ARTICLE BY THE CHIEF HYDROGRAPHER, ESPECIALLY TIMELY BECAUSE OF THE RECENT PUBLICATION OF SOME WILD REPORTS.

By R. Z. KIRKPATRICK.*

During the past twelve months several articles have appeared in semi-scientific and shipping magazines in which was mentioned a deficiency of Panama Canal water supply; the trend of the authors was to accept the assumed deficiency as a fact, after a misapplied discussion of some officially published figures; they then advocated schemes for converting the canal to sea level at admitted expenditures of decades of years and hundreds of millions of dollars. One writer pessimistically asserted that in May, 1920, the water supply was so deficient that house spigot leakages had to be prevented by the governor; of course, the writer mistook the intent of an official circular to prevent heedlessly wasting municipal water, filtered and pumped at considerable expense.

Perhaps a resume of the water supply situation of the Panama Canal, past, present and future, may help to clear up some misunderstandings among Americans honestly interested in a great future for the canal. It may be accepted as a fact that the supply has been and is being carefully studied and that the authorities have the status of the question well in hand; and that as the economics of the situation indicate the wisdom of more storage the same will be recommended to Congress, which no doubt will provide the funds.

ORIGINAL WATER SUPPLY ESTIMATES.

The present canal structures, dams, lakes, etc., were based on water supply estimates of the board of consulting engineers, who functioned in 1906. They estimated a minimum flow of 1250 c. f. s. into Gatun Lake during the three driest months of the year; and a minimum annual flow of 5730 c. f. s. The estimated minimum dry season flow was cut by them to 1000 c. f. s., however, for safety. From this was to be subtracted:

	Cubic Feet Per Second
Evaporation	710
Leakage at gates	240
Infiltration, waste at locks, etc.	77
Lighting, power, etc.	200
Total	1,227

Using 4 feet of storage (El. 86 to 82) and the 1000 c. f. s. of minimum three months' dry season inflow, they calculated the water supply would take care of the 1227 c. f. s. usages and provide 1350 c. f. s. for lockages, enough for 26 transits a day during the three driest months of the year. They estimated that when shipping required this amount it would be necessary to store more of the wet season water; and that their studies showed an Alhajuela reservoir would provide for 27 additional lockages; that the annual tonnage possible, both with and without Alhajuela reservoir was:

Tons per Lockage	Annual Tonnage without Alhajuela Dam	Annual Tonnage with Alhajuela Dam
3000	28,470,000	58,035,000
4000	37,960,000	75,380,000
5000	47,450,000	96,725,000

* Chief Hydrographer, The Panama Canal.

They added that should it eventually be necessary to provide for further tonnage, the Chagres and its tributaries would furnish additional water, which might be stored and subsequently utilized.

PRESENT WATER SUPPLY CONDITIONS.

It should be remembered that runoff conditions changed when Gatun Lake was formed. We have, therefore, started a new term of hydraulic records beginning in 1911. And our records are conservative, inasmuch as since 1911 we have had several dry years with but one of the abnormally wet ones, for instance as wet as in 1909 or 1910. There has been no actual shortage of water during this time. In the past two dry seasons, however, the storage between El. 87 and 80 has been drawn on considerably; in 1920 down to El. 81.65.

Usage for making power is the greatest. During the calendar years 1919 and 1920 power usages took 1470 and 1404 c. f. s. This is much higher than the consulting board anticipated; the development of many auxiliaries (such plants as the drydocks, pumps, shops and coaling plants), not included in the original canal project, is the cause. As a matter of fact the locks (expected by the consulting board to use the bulk of the power to be made) have actually taken but about 8½ per cent of all power made in the fiscal years 1918, 1919, 1920 and 1921. How little this is may be made clear by comparison with the power system losses in the transformers and transmission lines, which are about 16 per cent of the power made. It seems that greatly increased water usages for shipping will not necessarily be paralleled by like increased water usages for making power for the locks.

The actual inflow and usage of water from Gatun Lake in the calendar year 1920, the driest year since the operation of the canal, are below:

Inflow into Gatun Lake	Per Cent	Bill. Cu. Ft.
Runoff above Alhajuela	(38)	65.21
Yield from land area below Alhajuela	(44)	74.95
Direct rainfall on Gatun Lake	(18)	31.38
Total	(100)	171.54

The usage or disposition of this was:

	Per Cent	Bill. Cu. Ft.
Evaporation from lake	(13)	22.40
Gatun Lake lockages	(12)	20.87
Hydroelectric power	(26)	44.42
Spillway discharge	(47)	81.00
Leakage and municipal water	(1.5)	2.45
Increased storage	(0.5)	.40
Total	(100)	171.54

Nearly half of the year's water supply was wasted over the spillway—not needed. Yet the yield of water for the year was 12 per cent below the annual 1911-20 mean of 6183 cubic feet per second; and about 10,400,000 Panama Canal net tonnage transited the canal in 1920. Again, 26 per cent of the year's water was used for making power, and the Miraflores steam power plant was standing by, at two-thirds the hydroelectric plant's capacity, ready to be used if necessary.

FUTURE STORAGE DEVELOPMENT AND USAGE.

It has been seen that when a shortage of water does occur, due to increased business in the canal, it will be from lack of storage to tide over the needs of the dry season. That is, the 7 feet of storage between El. 87 and 80 in Gatun Lake will be insufficient pondage during the dry season, and since no more can be stored during the preceding wet season the excess must be wasted over the spillway. Careful records are kept of current hydrologies. When the stage mentioned above is reached, more and more power production will be switched to the Miraflores steam plant during these months. And the water that would otherwise be used in making power will be available for lockages.

When the economics of the situation warrant, storage developments can be constructed. By a relatively slight change at Pedro Miguel Lock in changing the system of drawing water, and improving navigation by cutting down surges in Gaillard Cut (at a cost of less than one million dollars), 1½ feet more over Gatun Lake becomes available as usable storage during the dry season—6.7 billion cubic feet more, or 985 extra lockages, may be taken care of, if all is used at the locks.

The fiscal year 1921 ratio of lockages to actual ships transiting the canal is 82 per cent; that is, many lockages are made two or more ships at a lockage. As the number of ships increases it will be possible so to dispatch shipping as to lower this ratio; that is, save much water by so-called tandem lockages. An example of such grouping occurred January 19, 1921, during the passage of the fleet. At Gatun Locks 18 lockages were made in passing 20 Navy ships, 8 commercial ships, 1 commercial launch, and 1 Panama Canal tug; 30 vessels in all.

When necessary the Alhajuela Lake and Dam can be constructed. By putting in a hydroelectric plant in the Alhajuela Dam (particularly with Miraflores steam plant in use), the power use of water can be entirely removed from Gatun Lake during the dry season and much water supplied for lockages in addition. Of course, the Alhajuela tail race water discharges into Gatun Lake, where it is again available for power or lockage use.

If further storage becomes necessary there are excellent possibilities in additional storages during the occasional super-wet years, carried over, if necessary, for several years. A dam in the Chagres arm of Gatun Lake just above the Gamboa Bridge can easily be built. There are also favorable reservoir sites in the Chagres above the proposed Alhajuela Lake. All these projects are along the ideas proposed originally for future construction by the board of consulting engineers, and it is believed their studies and estimates have proved as well advised as could be expected.—"Pacific Marine Review."

EIGHT-THOUSAND-TON MOTORSHIP WITHOUT STEERING ENGINE.

When one of the 8000-ton Diesel-powered freighters now building by the Deutsche Werft, Hamburg, for the Hamburg-American Line goes into service she will be unique among motorships, for she will have no steering-engine. She will be steered by a new type of rudder invented by Director Flettner, who recently demonstrated his device in the Hamburg shipbuilding trial tank. The first rudder of this type was installed about a year ago on the small Dutch steamer *Frigido* plying regularly between London and Rotterdam and has proven satisfactory under severe service, even in winter storms on the North Sea, and the vessel has navigated among the crowded

shipping of the River Thames with perfect control. The installation on the Hamburg-American Line's motorship will be the first on a large seagoing vessel, however, and the record of this new system will be closely watched.

No steam or electric steering-engine is required, the rudder being operated with slight expenditture of effort by a simple device which utilizes the pressure of the water passing same. At the aft end of the main rudder there is a small assistant rudder which is moved by a gear connected to the steering wheel in the pilot-house. By referring to the diagram this rudder will be

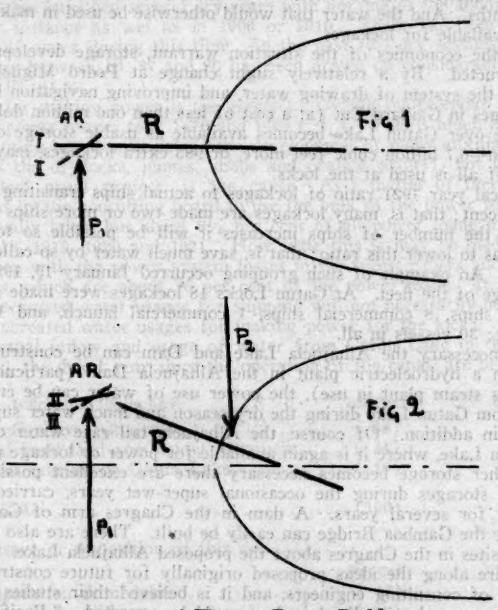


Diagram of Fleutner Patent Rudder.

noted at the extreme left and it will further be evident that if this small blade is moved from the longitudinal position I to the oblique position II that the pressure of the water P_1 acts upon it in the direction indicated by the arrow in Fig. I, which pressure moves the main rudder to a certain position as shown in Fig. 2 until the moment of the pressure P_1 is balanced by the moment of the pressure P_2 which acts upon the main rudder blade.

Further movement of the assistant rudder to III will be followed by further movement of the main rudder. The power required to turn the assistant rudder is quite small because the pressure P_2 is much greater than P_1 as its lever is much smaller, especially if the rudder is of the balanced type. If the assistant rudder is of the balanced type the pressure P_1 will serve for steering ships having an unbalanced rudder. In fact, it has been deter-

mined that only 5—20 per cent of the normal power is necessary to operate rudders on this system, the exact amount depending upon whether main and assistant rudders are of the balanced type or not. The gear needed to operate this rudder is small, simple and can be made very heavy to withstand all stresses coming upon it.—"Motorship," Aug., 1922.

CRYSTALLIZATION.

In spite of the progress of metallurgical knowledge and the wider spread which such knowledge is gradually receiving, one still hears frequent references to "crystallization" of metals as a result of exposure to vibration or fatigue stresses and as a cause of ultimate failure. The origin of this entirely erroneous theory is not at all easy to trace, unless it be that the association of brittleness with a strikingly crystalline character lies at the base of the idea. Such substances as crystalline sugar, rock crystal and even metals like bismuth or antimony exhibit at one and the same time a strikingly crystalline character and extreme brittleness and in some cases extreme fragility. Again, it sometimes happens that the fracture of a piece of iron or steel which has been broken by repeated blows or fatigue stresses exhibits visible crystalline facets, while a piece of externally similar material broken by gradual means, such as a tensile test, shows a fibrous or "tough" fracture. Thus, the view may have arisen that a good, tough, "fibrous" metal becomes converted into a weak brittle material having a markedly crystalline character by some process of "crystallization," which is brought about or favored by vibration or alternating stresses. The wide acceptance which this view has obtained, and the difficulty which is still to some extent experienced in substituting for it a view more in accordance with known facts, is sufficient evidence of the plausibility of the older theory. None the less, closer examination at once serves to show that it is entirely untenable.

The first misconception which lies at the base of the old idea of "crystallization" is that crystalline character and brittleness are inseparable. This is entirely untrue, and, indeed, is not only untrue, but the very opposite of the truth. We now know that all metals in their normal condition are crystalline, and that they owe their ductility or plasticity to their crystalline structure; so much so, that when, by such processes as cold working or quench hardening, we impair the perfection of the crystalline structure, we at the same time impair or destroy entirely their ductility. Pure lead is one of the most typically crystalline substances. This is, of course, in strong contrast with the typical brittleness of most mineral crystals, but it is now becoming clear that the brittleness of such substances is due, not to the fact that they are crystalline, but that the crystals are built up in a peculiar and complex way, arising from the fact that the substances are definite chemical compounds and not pure substances. Chemical compounds between metals are equally brittle in most cases, and the crystals of ordinary ductile metals owe their ductility to the fact that the atoms are arranged in a specially simple manner, which allows them to glide over each other in layers without becoming entirely detached. It is by interfering with this simple arrangement, as we do in severe cold working, that we destroy the ductility of the material. Now, setting aside the false assumption that crystalline character and brittleness are associated, we have to consider the different kinds of fracture which occur even in the same piece of metal under different conditions. Thus, a piece of good mild steel or wrought iron, if pulled out in a tensile testing machine, yields the well-known fibrous fracture. Here, again, one is liable to be misled by a false analogy. In a piece

of wood or cloth there is a true fiber, and its nature is well displayed by the appearance of a tensile fracture. But here the component particles or cells are not themselves able to undergo much deformation during the pulling process. Consequently, the structure of the material does not become changed during any elongating process, and the fiber one sees in the fracture is the true fiber of the material. In a ductile metal it is quite otherwise. There is, in the true sense, no fiber in any piece of metal in its normal condition, unless the elongated patches of non-metallic impurity can be thus described. But, during the elongation period of a tensile test, or in the corresponding phase of a bending test, the structure is plastically elongated —the individual crystals and whole groups of crystals are drawn out into elongated bodies very like fibers. It is these elongated crystals which finally break across at the fracture, and show us the "silky" or "fibrous" fracture—the exact appearance depending upon the size and arrangement of the original crystals. If, however, the same piece of metal is broken in another manner, so as to avoid any considerable drawing out, a totally different fracture results. Thus, if a bar of wrought iron or very mild steel is deeply notched, it can be made to fracture under a single heavy blow. Here the crystals break across along their cleavage planes and are not drawn out into pseudo-fibers, with the result that we find a brightly facetted "crystalline" fracture. The material was equally "crystalline" in both cases, but the kind of fracture developed in these two different ways is widely different, and the crystalline character is much more obvious in one than in the other.

Finally, the old theory of "crystallization" under vibration assumes that it is possible for metals like iron and steel to undergo internal rearrangement at the ordinary temperature. Actual observation shows clearly that this is not the case. Quite close to the fracture of a piece which has failed under fatigue or vibration the micro-structure is unchanged. It is true that the process of fatigue failure does bring about internal changes, but these are known to be changes occurring within the crystals of the metal, and not of the nature of any kind of crystalline rearrangement. In the case of nearly pure iron, re-crystallization does not occur at any temperatures below 500 degrees Cent., and even then only if the iron has been severely cold-worked. In softer metals, like gold, such changes can occur at rather lower temperatures, and in pure lead and tin they take place at the ordinary temperature of the air; but in all cases such changes render the metal softer and more ductile, not harder and more brittle. On every count, therefore, the old theory of "crystallization" must be dismissed as untenable, and although the whole of the phenomena of fatigue failure are not yet elucidated, a better explanation is available. In the simplest terms this is that fatigue failure results from the fact that a stress which is unable to produce measurable permanent deformation when steadily applied, may yet be able to cause a slight amount of movement or slip in a few of the crystals of a piece of metal, and if this movement is repeated, or, worse still, reversed, many thousands of times it brings about the disintegration of the material in its immediate vicinity. This, sooner or later, leads to further movement and subsequent disintegration in adjacent crystals, until a fracture is formed. Although at first sight less simple than the old theory, this view leads to many important practical conclusions. We can refer to only one of these. The old theory led engineers to adopt, as a precautionary measure, the practice of having parts exposed to vibration or alternating stress periodically "annealed," and this practice is still followed in a number of cases. The object was, of course, to undo the "crystallization" which was supposed to have been taking place. What

is its actual effect? Here everything will depend upon the exact conditions. If fatigue has progressed so far that even a few of the crystals have already become fractured, nothing can repair the damage, and annealing is liable to do more harm than good. Also, if the part is strong enough not to undergo fatigue failure in continued use, then, again, annealing is likely to do harm. The reason here lies in the fact that unless an annealing operation is conducted with great care and knowledge, it tends to leave the metal with a lower elastic range than it had in the original condition, and such treatment inevitably tends to shorten rather than to lengthen the life of a part exposed to fatigue or vibration. An important example is the practice of annealing the cylinders used as containers for high-pressure gases—a practice which is certainly not consistent with modern metallurgical knowledge.—"The Engineer," Aug. 25, 1922.

THE INTERNAL COMBUSTION TURBINE.

Including a Description of a Proposed New System of Operation.

By W. A. D. Forbes, R. C. N. C., A. M. I. N. A.

In a recent number of the *Revue Général de l' Electricité*—February 18, 1922—there appeared an interesting and informative article by M. Jules Deschamps on "The Gas and Oil Turbine," in which he discusses a paper on the subject which was read in Germany by Professor Schüle in May, 1921, and which was published in the *Elektrotechnische Zeitschrift* on July 21 and 28, 1921. Professor Schüle gives in his paper particulars of the Holzwarth constant volume type of turbine, and of the trials recently carried out, and amplifies the account given by H. Holzwarth himself in his paper read in 1920 and published in the *Zeitschrift d. Vereins d. Ingenieure*, and also gives plans of designs of such turbines having an output of 10,000 kilowatts.

M. Deschamps challenges Professor Schüle's conclusion that the constant-pressure type of gas turbine cannot succeed, and that the explosion type of Holzwarth alone has a future, and maintains, on the contrary, that the outlook for the constant-pressure type is very favorable.

Comparison of Types and Efficiencies.

M. Deschamps contends that in seeking for a solution of the gas turbine problem it is necessary continually to bear in mind the historical development of the piston type, since the principles of thermo-dynamics are invariable and apply equally to both the turbine and the piston engines, so that the study of the development of the one should afford a guide to that of the other. He points out that to increase the efficiency of the early piston engine the compression was gradually raised until the limit was reached, beyond which it was not possible to go without danger of premature explosion. To increase still further the efficiency by raising the compression it became necessary to inject the fuel at the end of the compression stroke, thus leading to the Diesel type of constant-pressure cycle. The same reasoning holds good for the turbine types and indicates, as the criterion of success, the employment of high compressions and the burning of the fuel under steady conditions.

The single-fluid constant pressure turbine has the merit of simplicity, and no complicated mechanically operated valves exposed to hot gases are required. It can be worked on a sub-atmospheric cycle, and the burning of the fuel presents no difficulties, as it can be burned at atmospheric pressure

in a combustion chamber lined with refractory material. The sub-atmospheric turbine possesses other important advantages; extreme lightness of construction is possible, as the maximum pressure never exceeds 14 pounds per square inch. It is also possible to cool the working parts by flushing with cold air, which can afterwards be used in the combustion chamber; also, air "leaks in" can be provided for cooling the blade roots with but slight loss of efficiency. Moreover, owing to the lower density, the frictional losses in the turbine, the cooler and the pump are greatly reduced, thus leading to a substantial improvement of efficiency. The temperature and velocity conditions are very steady throughout the cycle, and there are no sudden fluctuations such as occur in the constant-volume cycle—an important consideration, since the motor blades are subjected to high centrifugal stresses. The chief disadvantage of the cycle is that the negative work forms a comparatively large part of the total, and a big compressor is therefore necessary. Unfortunately, a piston compressor, which has a high efficiency, is out of the question on account of the bulk of the gases to be dealt with, while present-day rotary compressors have a poor efficiency if anything beyond a moderate compression is required. In the mixed-fluid constant-pressure turbine, the ratio of positive to negative work is greater than is the case in the single-fluid type, as the steam used may be condensed. The compressor required is not so big and the importance of its efficiency diminishes. The combustion chamber can be surrounded by a steam boiler which enables the radiation losses to be reduced. In dealing with the question of steam injection for diluting the working substance and lowering its temperature, M. Deschamps criticises Professor Schüle's contention that the lowering of the temperature in the Armengaud-Lemale turbine by steam injection was fatal to its efficiency, and calls attention to a paper by M. Sekutowicz before the Société des Ingénieurs Civils de France in 1906, in which it was shown in a striking manner that the injection of steam into a combustion chamber does not have any material effect on the efficiency, and permits certain advantages to be secured. This particular point has been very adequately dealt with by Mr. Norman Davey in his excellent book on the gas turbine, and Professor Schüle would appear to be at fault on this point. As M. Deschamps remarks, the trials of the Armengaud-Lemale turbine gave much interesting information, its defects have been recognized and much better results could now be obtained when operating on this type of cycle. The great objection to the mixed-fluid type is that, on account of a small percentage of sulphur which is present in the fuel, corrosion of the metal parts will most likely take place.

The constant-volume turbine is less dependent upon the compressor, as in this cycle the ratio of positive to negative work is much larger than in the constant-pressure type. The constant-volume type has the disadvantage of being more complicated owing to the mechanically operated valves subjected to high temperatures which are required to close the combustion chamber while the explosions are taking place. The pressure in the combustion chamber is continuously falling during the expansion, so that the velocity with which the gases leave the nozzles and enter the moving blades must be continually diminishing, a state of affairs that is not conducive to efficiency. Both Professor Schüle and H. Holzwarth, however, claim that the varying velocity does not materially diminish the efficiency. The gas flow through the turbine wheel is considerably different in the constant-volume turbine from what is the case in either the constant-pressure or steam turbines. For the latter the gas or steam flow is steady, whilst in the constant-volume turbine the combustion products are ejected from the combustion chambers like machine-gun fire, the pulses of the jets of gas being intermittent, of

short duration, and acting like a hammer blow, as the emptying of the chamber and the flow through the wheel does not take more than one-tenth to one-twentieth of a second. The stresses on the blades are, as a consequence, much greater than is the case with steady flow. A further disadvantage is that it is necessary to inject cold air for scavenging and to cool the combustion chamber walls and turbine blades; this air, which has only a small pre-compression, will therefore leave the nozzles and pass through the blades with a much lower velocity than the hot gases, which again is not conducive to efficiency. It becomes questionable as to whether it is better to inject the cold air separately or mix it with the products of combustion.

The Holzwarth turbine does not lend itself readily to a sub-atmospheric cycle, and therefore suffers a disadvantage in that the frictional losses in the turbine are enhanced and the mechanical efficiency thereby reduced. A good overall efficiency can be obtained by working on a cycle of comparatively low thermal efficiency with small mechanical losses, or by working on a cycle of higher thermal efficiency with larger mechanical losses. The turbine has a lower mechanical efficiency than the reciprocating engine, and to obtain the same, or greater, overall efficiency it is necessary for the turbine to work on a cycle of greater thermal efficiency, and it is only by making use of a better vacuum that the steam turbine has been able to rival the reciprocating engine. The Diesel engine working on a super-atmospheric cycle owes its high efficiency to the fact that the operations of expansion and compression are both carried out in this type of mechanism with great efficiency, thus giving the engine, as a whole, a high mechanical efficiency. In order to obtain an efficiency approaching that of the Diesel engine it will be necessary to work an internal combustion turbine on a sub-atmospheric cycle, and the constant-pressure type alone is suitable for such operation.

Professor Schüle gives curves in which he compares the efficiencies of the Holzwarth and constant-pressure turbines with a piston engine working on an "Otto" cycle. The diagrams given in his paper, in which thermal efficiency is plotted to a base of compression ratio, show that the curves for the constant-pressure turbine and the piston engine lie very close together, while that for the Holzwarth turbine lies distinctly above them, particularly at low compressions. From these curves Professor Schüle concludes that the efficiency of the Holzwarth turbine is greater than that of the piston engine and constant-pressure turbine at all values of the compression ratio, and particularly so for low compressions. He omits, however, to mention that the constant-volume turbine can only operate with a low pre-compression, while the constant-pressure turbine can utilize a much greater compression ratio, and that a fair comparison is not given by comparing the efficiencies obtainable at the same compression ratio; moreover, the thermal efficiency alone does not represent the whole story.

M. Deschamps remarks that the efficiency of the Holzwarth turbine which functions with a feeble pre-compression is insufficient, and that the trials have shown all that it is capable of. He hails with satisfaction the statement of Professor Schüle that its efficiency is greater than that of a piston engine of the same compression ratio as showing that the more complete expansion and diminished heat losses to the walls surpasses the disadvantage of the turbine's lower mechanical efficiency. It is difficult, however, to believe this claim, and in the absence of definite statements as to the fuel consumption per shaft horsepower available for external work, the writer considers that such a claim cannot be definitely established, and in any case such an efficiency would not be sufficient to enable the gas turbine to compete with the steam turbine.

Finally, Professor Schüle remarks that, although with the 1000 horsepower trial gas turbine good efficiencies up to 1200 horsepower were shown on the wheel circumference, the difficulty of transmitting the greatest possible efficiency to the shaft still remains, but that with the further developments of the experimental work which was contemplated there was good ground for expecting that at no distant date the Holzwarth turbine would reach as high a degree of efficiency as the steam turbine.

THE HEPBURN-FORBES INTERNAL COMBUSTION TURBINE SYSTEM.

As a result of some years of study of the problems involved, the writer of this article and his colleague, Mr. H. A. Hepburn, B. Sc., A. M. I., Mech. E., have proposed a new system of operation, a description of which it is hoped may be of interest, as it is based upon a new theory of nozzle action and involves the use of a novel type of pump, known as a "kinetic compressor."

In the Hepburn-Forbes system the sub-atmospheric, single-fluid, constant-pressure cycle has been adopted, as it is considered that this type of cycle yields the most promising line of development.

Attention has been directed towards the employment of greater compression ratios than are possible with a rotary compressor, and the utilization of the kinetic energy remaining in the gases after leaving the moving turbine blades. To attain these ends it is proposed to use a kinetic compressor, in which the kinetic energy of the gases is converted into pressure energy. How this is to be done will be described later. It can readily be seen from theoretical considerations that an increase in the mechanical efficiency of the expansion part of the cycle will lead to a substantial improvement in the overall efficiency, and that this is particularly the case for the constant-pressure type of turbine. By allowing the gases to leave the turbine with sufficient velocity to enable the whole of the compression work to be carried out in a kinetic compressor it is found that there is an improvement in the overall efficiency which may amount to the order of 80 per cent or more above that obtained when a rotary compressor is employed.

Having given up the rotary compressor, it becomes necessary to produce and maintain the vacuum by other means. It is proposed to maintain the vacuum by the expansion of high-pressure gas in a nozzle system. When gas expands through a nozzle there is a tendency in certain circumstances to go on expanding beyond the back pressure. What vacuum pressure can be produced by the expansion of gas through a nozzle from a pressure greater than that of the atmosphere? The theory advanced by Mr. Hepburn and the writer is, that under favorable circumstances the expansion will continue until the atmospheric line becomes the M. E. P. line of the P. V. diagram. That is, if p_0 in Fig. 1 is the initial pressure and p_1 the atmospheric pressure, then the expansion would continue until a pressure p_2 was reached, such that the shaded area A B C F was equal to the shaded area C E D.

So far as the writer is aware, this particular relationship has not hitherto been published by others. Experimental evidence confirming this theory can be obtained from the nozzle experiments of Stodola. On page 64 of his book on "Steam Turbines"—second English edition—Stodola gives a series of curves obtained with a diverging nozzle, using steam, in which the back pressure was given various values greater than that for which the nozzle was designed. After passing the minimum section the pressure fell along the line of adiabatic expansion until a point was reached considerably below the value of the back pressure, and afterwards rose until the back

pressure was reached. Assuming the above theory to be true, it will be found that the value of the minimum pressure so calculated is in each case in fair agreement with the experimental result. The theory here advanced explains many interesting points concerning nozzles and orifices and the trains of waves which sometimes arise, and also points in the action of ejectors, such as the reason why a considerable drop in the initial pressure gives only a slight decrease of vacuum. It would take up too much space,

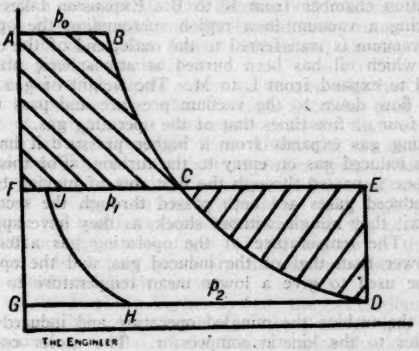


FIG. 1

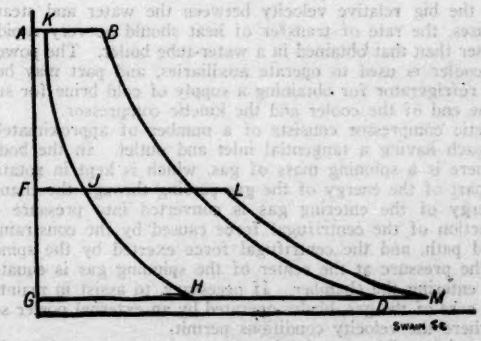


FIG. 2

however, to enter more fully into this matter, but it is suggested that it would form a starting-point for formulating a theory of the action of an ejector.

When the vacuum has been established a further supply of air, in which fuel has been burned at atmospheric pressure, can be induced to expand down to the vacuum pressure, and do work in the turbine. The gas which expands from the pressure p_0 is termed the "operating gas," and the gas

which it induces to flow from the atmospheric pressure p_1 down to the vacuum pressure p_2 is termed the "induced gas." The gases, after doing work in the turbine, are cooled at constant pressure to the point H—Fig. 1—and are then compressed in the kinetic compressor back to atmospheric pressure. The amount of induced gas which enters the system per pound of operating gas depends upon the ratio of the areas A B D G and F J H G.

The sequence of operations is as follows:—Operating air is compressed in a mechanical compressor from J to K—Fig. 2. Oil is burned in a high-pressure combustion chamber from K to B. Expansion takes place from B to D, producing a vacuum in a region surrounding the operating gas nozzle, and the vacuum is transferred to the outlet end of the cooler. The induced gas in which oil has been burned at atmospheric pressure from J to L, is caused to expand from L to M. The weight of gas that can be thus induced to flow down to the vacuum pressure and pass through the turbine is about four or five times that of the operating gas.

As the operating gas expands from a higher pressure it has a greater velocity than the induced gas on entry to the turbine. For this reason the operating gas alone is passed through the first row of moving blades. The operating and induced gases are both passed through the second row of blades, and on exit they mingle without shock, as they have approximately equal velocities. The temperature of the operating gas after expansion may be much lower than that of the induced gas, and the operating gas may therefore be used to give a lower mean temperature to the second row of blades.

On exit from the turbine the mingled operating and induced gases pass through the cooler to the kinetic compressor. The cooler consists of a steam boiler in which the water and steam flow in a direction opposite to that of the exhaust gases. The cooler is so arranged as to permit of the exhaust gases flowing through with a minimum reduction of their velocity. Owing to the big relative velocity between the water and steam and the exhaust gases, the rate of transfer of heat should be very rapid and very much greater than that obtained in a water-tube boiler. The power obtained from the cooler is used to operate auxiliaries, and part may be used for working a refrigerator for obtaining a supply of cold brine for surrounding the extreme end of the cooler and the kinetic compressor.

The kinetic compressor consists of a number of approximately circular chambers each having a tangential inlet and outlet. In the body of each chamber there is a spinning mass of gas, which is kept in rotation by absorbing a part of the energy of the gas passing through the chamber. The kinetic energy of the entering gas is converted into pressure energy by the joint action of the centrifugal force caused by the constrained motion in a curved path, and the centrifugal force exerted by the spinning mass of gas. The pressure at the center of the spinning gas is equated to that of the gas entering the chamber. If necessary, to assist in maintaining the spin of the core of the gas, blades operated by an external power source may be fitted where the velocity conditions permit.

Whether it is possible to maintain such a spinning mass of gas is a matter for experiment. If it is feasible, the writer feels confident that a solution of the gas turbine problem has been achieved. In a paper on "Flow of Air Through Nozzles," by Morley, which was read before the Institution of Mechanical Engineers in 1916, it was shown that when a jet is discharged into a room, the air in the room flows into the jet, and vortices are set up, which, once set going, continue without abstracting any appreciable quantity of energy from the jet. In the suggested kinetic compressor the kinetic energy of the spinning gas is a very small fraction of that of the gas which passes through the compressor.

If successful in action, a pump of this type has great advantages over a rotary compressor; there is, for instance no leakage loss and no bearing friction loss, and it would be possible to utilize much greater compression ratios. The bulk of the pump will be very greatly reduced on account of the much higher peripheral velocities employed, and it will be possible to obtain in three stages the same compression ratio which would require more than ten stages in a rotary compressor. Moreover, the simplicity of the compressor makes it exceedingly attractive. In the kinetic compressor the gas would be in a constant state of deceleration, whereas in a rotary compressor the gas is accelerated and decelerated many times in the course of compression, with consequent loss in eddies and shocks, apart from the intricacy of the path of motion.

The natural objection which occurs to anyone who has not given much consideration to the matter, is that the high velocities used will lead to big frictional losses, but a closer investigation shows that, contrary to what would be expected at first sight, these losses are not excessive. This is chiefly due to working on a sub-atmospheric cycle, which gives a low average density, and also the high velocities lead to small surface areas and a small number of stages in the compressor. The high gas speeds used should facilitate the rapid transmission of heat to the cooling medium.

One of the principal advantages of the system is that, apart from auxiliaries, the only moving parts are the turbine wheels and the gas, a simplicity it would be difficult to rival.

Operating on the Hepburn-Forbes system, the calculations, in which reasonable allowances have been made for losses show that an overall efficiency approximating to that of the Diesel engine should be obtained, and there is therefore a wide margin for unforeseen contingencies between this value and the efficiency of the very best steam turbine.—"The Engineer," Sept. 1, 1922.

BOOK REVIEW.

NOTES AND SKETCHES ON MARINE DIESEL OIL ENGINES. A Manual of Marine Oil Engine Practice by J. W. M. SOTHERN, D. VANNOSTRAND CO., 8 Warren St., New York, N. Y. Price, $8.00.

A book for which marine engineers have been waiting for years. It is written in the usual clear style the author uses in text books. It is thoroughly up-to-date, which in this day of rapid progress in Diesel engines is quite an accomplishment. It contains a description of the principal types of marine Diesel engines, methods of operation and upkeep, and useful information including a short treatise on liquid fuels, various auxiliary appliances, and a very complete description of the indicator diagram.

MACHINERY AND PIPE ARRANGEMENTS ON SHIPBOARD, by C. C. PORMDER. EMMOTT & CO., LTD., 65 King St., London, England. Price, $6.00.

A very useful book for the engineer of the Merchant Service. It gives the latest practice in regard to the arrangement of the main and auxiliary machinery and various piping systems, including forced lubrication systems as developed for turbine work. It also gives much data that is useful to the designer of this type of machinery. The book is very clearly written and the subject matter is concisely presented.

OBITUARY.

CHIEF ENGINEER FREDERICK G. McKEAN, U. S. N.

Chief Engineer Frederick G. McKean, U. S. N., retired, died on August 2 in Washington, D. C., where he had resided off and on since 1863. He was born in Bombay, East Indies, in 1838, but came to the United States in 1851; went to the Grammar School of St. James in Maryland three years and to the preparatory class of the Rensselaer Polytechnic Institute in Troy, N. Y.; served his time at the Vulcan Works of Murray & Haylehurst, in Baltimore, and entered the engineer corps of the United States Navy in February, 1861. During the Civil War he served in the *Mississippi, Estrella, Ascutney* and *Canonicus*. After the war he served on board the *Ticonderoga, Ashuelot, Ticonderoga* again, *Marion* and the *Boston*, from which vessel he was retired in 1893 for partial deafness. During his active service he was attached to the bureau of steam engineering, Navy Department, five different times, and was on other duty a short time during the war with Spain. He was a member of the Protestant Episcopal Church, also a Mason, Knight of Honor and belonged to the Military Order of the Loyal Legion of the United States and the AMERICAN SOCIETY of NAVAL ENGINEERS. He was of a retiring disposition, was an indefatigable reader and wrote a few short articles on various subjects. He married a Philadelphia lady, who survives him, and he leaves two sons—Fred G. and Robert C., both lawyers.

ASSOCIATION NOTES.

The Annual meeting of the Society for nomination of officers for the next calendar year was held at the Navy Department, Washington, D. C., on Tuesday, October 3, 1922. The following were nominated:

For *President:*
Captain J. T. Tompkins, U. S. N.

For *Secretary-Treasurer:*
Commander S. M. Robinson, U. S. N.

For *Naval Member of Council:*
Rear Admiral B. C. Bryan, U. S. N.
Captain D. E. Theleen, U. S. N.
Captain Robert Stocker (CC.), U. S. N.
Captain P. B. Dungan, U. S. N.
Captain H. L. Wyman, U. S. N.
Commander H. S. Howard (CC.), U. S. N.
Lieut. Commander A. M. Charlton, U. S. N.
Engineer-in-Chier Q. B. Newman, U. S. C. G.

For *Civil Member of Council:*
Mr. C. F. Bailey.
Mr. Robert Warriner.
Dr. Edgar Buckingham.
Mr. Elmer P. Rust.

It was decided to place before the membership the matter of holding a dinner in 1923, and provision has been made on the ballot for officers for a vote on this matter.

Through the courtesy of the American Marine Association, the Society was represented in the American Marine Week in Grand Central Palace, New York City, November 4 to 11, 1922. This was a most successful event. A large number of members were greeted and it is believed that the Society made many new friends.

The following members have joined the Society since the publication of the last JOURNAL.

NAVAL.

Clark, Joseph A., Ensign, U. S. N.
Donelson, John F., Lieutenant Commander, U. S. N.
Harrigan, Daniel W., Ensign, U. S. N.
Heller, Hyman L., Ensign, U. S. N.
Ker, Lucien J., Lieutenant (E), U. S. C. G.
Lester, John C., Ensign, U. S. N.
Martin, James A., Ensign, U. S. N.
Mitchell, Mason E., Lieutenant (SC.), U. S. N.
Moses, Moultre, Ensign, U. S. N.
Pape, William B., Ensign, U. S. N.
Scofield, William R., Lieutenant, U. S. N.
Settle, Thomas G. W., Lieutenant, U. S. N.
Shipley, Walter M., Ensign, U. S. N.
Smith, Arthur A., Lieutenant, U. S. N.
Smith, Robert Mc L., Lieutenant, U. S. N.
Ten Eyck, John C., Jr., Ensign, U. S. N.
Thornhill, Henry E., Lieutenant, U. S. N.
Zimmerman, Walter E., Ensign, U. S. N.

CIVIL.

Kerschner, Edward, Wilkins Building, Washington, D. C.
Warriner, Robert, Chief Engineer, Bethlehem Shipbuilding Corporation, Quincy, Mass.

ASSOCIATE.

Clausen, H., Engineer-Lieutenant, Royal Danish Navy.

In addition to the above changes, the following have been transferred from Associate Membership to the new grade of Civil Member.

Bailey, Charles F., Engineering Director, Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

Buckingham, Edgar, Ph.D, Bureau of Standards, Washington, D. C.

Cheyney, Algernon R., Consulting Engineer, Mills Building, Washington, D. C.

Kahrs, Otto, Engineer, Raadhusgaten 1 & 3, Kristiania, Norway.

Kothny, G. L., P. O. Box No. 60, Strafford, Pa.

Leahy, Francis W., President of The Leahy Engineering Co., 709 2d Nat. Bank Bldg., Cincinnati, Ohio.

Lowe, Major Alonza W., 1 Olive St., Lynn, Mass.

McAdam, Dunlap J., Jr., Metallurgist, Engineering Experiment Station, Annapolis, Md.

Parsons, H. De B., Consulting Engineer, 22 William St., New York City.

Rust, Elmer P., Asst. Mgr. Operation Dept., United States Shipping Board, Washington, D. C.

Seward, Herbert L., Professor, Sheffield Scientific School, Yale University, New Haven, Conn.

Slocum, Stephen E., Consulting Engineer, General Machinery Foundations Co., Harrison Building, Philadelphia, Pa.

Warrington, James N., 1820 Taft Ave., Los Angeles, Calf.

ADDRESSES.—PLEASE DO NOT NEGLECT TO KEEP THE SECRETARY ADVISED OF YOUR ADDRESS AT ALL TIMES.